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DIESEL OPERATION AND MAINTENANCE

by

ORVILLE L. ADAMS, SR.

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London: 1947

CHAPMAN & HALL, LTD.

37 Essex Street, W.C.2

New York: 1946

PRENTICE-HALL, INC.

PRINTED IN THE UNITED STATES OF AMERICA

To My Wife

RHEBY LOUISE ADAMS

Whose Unbounded Faith Was an Inspiration

PREFACE

It is hoped that this book will be useful to anyone concerned with the operation and maintenance of Diesel engines. The practical arrangement of the material should make it especially valuable to those who have had Diesel engine experience. However, the scientific and technical problems involved are discussed in such a manner that they should be easily understood by students in technical institutes, vocational schools, and trade courses.

The first five chapters explain the basic principles and procedures of engine operation, identify the major problems, relate the origin of all practical difficulties to the fundamentals, and point out the limitations of the design and application of the Diesel. The metallurgical problems, the basic problems of operation, and the fuel-mixing problems are presented in their relation to maintenance.

Another five chapters cover maintenance and repair procedures, inspection methods, and techniques relative to piston and liner problems, ring problems, and bearing problems. Complete instructions are given for diagnosing all engine difficulties with reference to failures and breakage of engine parts, fuels and combustion, and lubrication. Check lists of engine troubles presented in several chapters serve as trouble-shooting guides to help the operator determine the causes of engine difficulties and the nature of the problems involved.

The purpose is to explain how the operator will encounter these difficulties, how to reduce them to a minimum, and how to keep practically all of them under control through the practice of preventive maintenance, the use of good standard operating practice and technique, and skillful maintenance methods. At the end of each chapter questions emphasizing the salient points of the subject and references to additional sources of information are given.

All the problems of Diesel operation and maintenance are inherent in the fundamental principles. The origin of such problems is explained; certain theoretical considerations and scientific reasons for the existence of the basic problems are discussed and illustrated. Other limitations imposed upon the engine by design and metallurgical considerations, which are related to maintenance and operation, are fully presented. Some problem solutions required years of effort by the designer as he coped with the inherent difficulties in applying the Diesel engine to present-day uses. While the designer has not yet solved all the problems of fuel, fuel mixing, and lubrication, understanding how to handle these operating difficulties greatly helps the operator to get the most service and economy out of his engine.

The material in this book grew out of a teaching experience. While on duty with the Navy in Alaska, the author was Ship Repair Officer in charge of the maintenance and repair of a fleet of Diesel-powered naval auxiliary ships, mine sweepers, tugs, and patrol craft, and experienced all kinds of operation and maintenance troubles. In addition, he taught Diesel maintenance and operation to several classes of Army and Navy personnel who were using their own time to prepare for postwar employment in this growing field. This experience with several classes over a period of two years indicated the need for certain technical information by the students if they were to obtain a satisfactory grasp of the problems involved, and also indicated that this required a study of the problems themselves rather than engine details.

Accordingly, available material for instruction was analyzed to determine what scientific and technical information should be correlated with the principles, methods, and techniques of operation and maintenance. This specialized information was then classified according to its relation to major problems, organized for instructional purposes, and incorporated in a series of lectures with illustrations. The material and instruction methods so developed are now presented in this book in a manner intended to be suitable for home-study and classroom instruction.

ORVILLE L. ADAMS, SR.

ACKNOWLEDGMENTS

THE author sincerely acknowledges the assistance and contributions of a number of individuals, manufacturers, and engineering societies received in the course of the preparation of this book.

Some of the material is abstracted from various papers and proceedings of the Society of Automotive Engineers, the American Society of Mechanical Engineers, and papers describing the work of research engineers released by the manufacturers. Whenever possible, mention of the source of the material has been made in the text. Special permission to use illustrations and quote from the manufacturer's material is acknowledged. In abbreviating such material, every effort has been made to avoid distortion of the original information in any manner, and a minimum of such material was used.

Special thanks and appreciation is extended to the author's foster son, Technical Sergeant Richard C. King of the Army Air Forces, for his help in making a considerable number of the original drawings and graphs. The technical editorial assistance rendered by Miss Mary L. Moran in the preparation of the manuscript and reading the proofs is a distinct contribution to this work.

The author was associated with Commander Charles L. Holley, USNR, for a year at the Navy Department, Washington, D. C., in the development of the Navy's post-war apprentice-training program. Commander Holley was the source of much helpful advice and suggestions on the vocational aspects and objectives of technical and trade training pertinent to the plan of the book.

While serving as the author's staff of junior officers in Alaska, Lt. William F. Richardson, Mr. Charles F. Kennedy, Mr. Leroy Dees, and Mr. Charles Bradburn supplied many practical pointers on overhaul and maintenance and assisted

in the development of instruction material incorporated in various parts of the book.

The author received and made use of considerable instruction material supplied by Dr. P. H. Schweitzer, Professor of Engineering Research, Pennsylvania State College, which he had developed and used for training several hundred Navy officers in Diesel engineering during the war. This material was most satisfactory for training purposes, and very useful in helping the author develop the technique of presenting such information in lectures.

The following manufacturers and publications provided many of the photographs used for illustrations: American Bearing Corporation, St. Louis, Mo.; Bacharach Industrial Instrument Company, Pittsburgh, Pa.; Briggs Clarifier Company, Washington, D.C.; Fairbanks Morse & Company, Chicago, Ill.; The Guiberson Corporation, Dallas, Texas; The Guiberson Diesel Engine Company, Dallas, Texas; Lisle Corporation, Clarinda, Iowa; Joshua Hendy Iron Works, Sunnyvale, California; *Petroleum Engineer*, Dallas, Texas; *Southern Power and Industry*, Atlanta, Ga.; *Diesel Progress*, New York, N.Y.; and Cooper-Bessemer Corp., Mt. Vernon, Ohio. Additional sources are mentioned in the text or with the illustrations.

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DIESEL OPERATION
AND MAINTENANCE

CHAPTER 1

FUNDAMENTAL PROBLEMS

Introduction. Diesel engines, in recent years, have been adapted to many new and severe uses. Inherent problems of operation and maintenance had to be studied and solved. As a result, various changes have been made in engine operation, in maintenance, and in the design of pistons, rings, cylinder liners, valves, and other operative parts. Continued progress and the development of new techniques promise that the immediate future will doubtless witness further improvements in Diesel engine design, and these should eliminate, to a great extent, most of the remaining objectionable features of operation.

The proper approach to the study of the Diesel involves certain definite considerations, which from the outset must answer two important questions:

1. What are the advantages of the Diesel inherent in its design and principle of operation? What technical developments have overcome its early disadvantages?

2. What are the limitations, the inherent disadvantages, if any, that have a bearing on its operation and maintenance and that must be considered with regard to its future development?

The purpose of this chapter is to present, in simplified terms, the answers to these questions. The basic characteristics of the Diesel engine, how these compare with those of other prime movers, such as the gas turbine and the gasoline engine, and how they affect future engine development, with related problems of operation and maintenance, should be understood.

Advantages of the Diesel engine. Consider first the advantages of the Diesel engine. These are not far to seek:

1. Fuel economy. The fuel economy of the Diesel engine is its outstanding advantage over the Otto cycle engine or the gas turbine. The difference is almost always more than 20 per cent on the pounds-per-brake-horsepower basis, and it frequently

runs much higher. To this fact must be added that Diesel fuel costs less than gasoline. Then, as far as fuel is concerned, horsepower is produced by the Diesel at a lower cost than by the

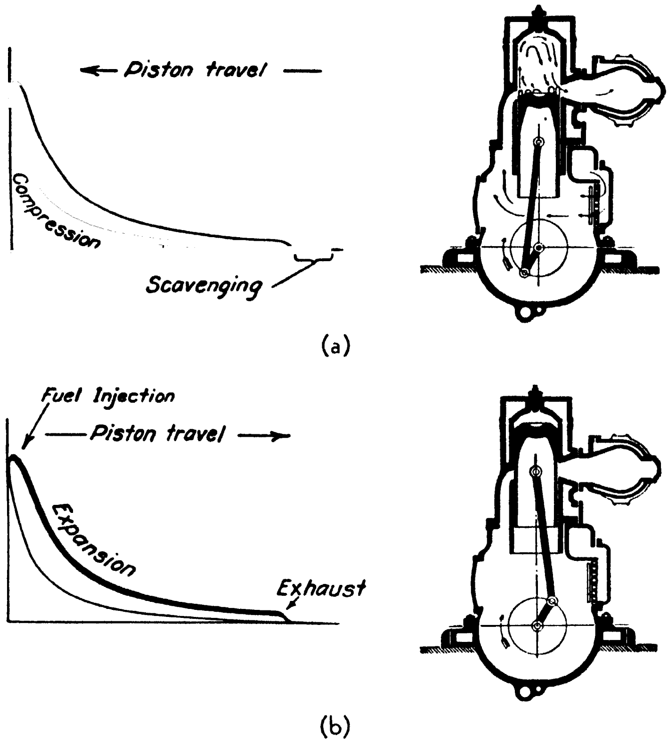


FIG. 1-1. (a) In the position of the piston shown in the top diagram, fuel injection has taken place and combustion has begun. The piston then moves downward and first uncovers a portion of the exhaust ports. Burned gases are exhausted to the atmosphere. When the piston has moved downward a small distance farther the air inlet ports are uncovered, and air from the crankcase under a slight pressure then flows through the cylinder, clearing out the remainder of the burned gases and filling the cylinder with fresh air. (b) As the piston moves upward on the compression stroke the air inlet and exhaust ports are covered and the entrapped air is compressed. Just before the piston reaches the top dead center, fuel oil is injected into the combustion space and ignition begins as the piston reaches the top of its stroke. The cycle has then been completed with the finish of the second stroke; hence the name *two-stroke cycle* or *two-cycle*.

gasoline engine. An equally outstanding advantage is that, at part loads, the Diesel engine is much more economical per horsepower hour than a gasoline engine of the same size. Although the Diesel engine operates better when the fuel variations are

held within the limits for which it is designed, it burns a wider variety of fuels more successfully than does the gasoline engine.

2. Efficiency. The Diesel engine is the most efficient of all internal combustion engines, owing to the fact that it employs the most direct method of transforming heat energy of combustion to mechanical energy that has been discovered by man. The reason for this will become apparent as we study the data presented in this book.

3. Fire hazard. There is much less fire hazard with Diesel than with gasoline fuel, for the Diesel fuel is not highly inflammable. In many applications, such as marine, stationary uses, and in the airplane, this advantage is not only desirable but essential.

4. Maintenance troubles. The ignition system and the carburetor are two important sources of maintenance trouble in the gasoline engine that are entirely absent from the Diesel engine.

5. Fuel distribution and burning. A more uniform fuel distribution, with consequent uniform combustion of the fuel mixture, is realized in the Diesel engine than in other engines; the fuel is thus metered in equal amounts to each cylinder by the fuel injection system.

6. Lugging ability. On account of the slower burning of the fuel oil in the Diesel engine, there is a much longer period of expansion of the gases. This provides greater lugging ability than the gasoline engine.

Disadvantages of the Diesel. There are still very definite disadvantages of the Diesel engine. Among these are:

1. Higher cost of installation. The Diesel engine naturally costs more to buy and install. This is due to the fact that it must have a precision-built fuel injection system, and also to its greater weight per brake horsepower. One of the chief reasons for higher costs is that a smaller number of engines are built, since the demand for such engines does not as yet justify tooling the factories for mass production as is the case with gasoline engines.

2. Diesels weigh more. The automotive Diesel is usually heavier than a comparable gasoline engine, mainly because it has to be sturdier and heavier to withstand the high stresses imposed during its operation by the high compression and explosion pressures. The design of such engines must provide satisfactory factors of safety for the stresses imposed when full power is developed. Such heavy parts as pistons, rods, and

shafts place a definite limit upon the revolutions per minute, and hence the power output.

3. Noisy operation. The majority of Diesels are definitely noisy and frequently very rough in operation. This is due to high peak pressures and characteristics of combustion in the cylinder.

4. Smoky exhaust. A very objectionable difficulty that has existed since the early days of the Diesel has been the smoky exhaust. The smoke has been greatly reduced, but it is still no minor factor where smoke is objectionable. More efficient fuel injection and improved combustion should continue to reduce the amount of smoke in the exhaust.

5. Starting difficulties. The Diesel will always be more difficult to start than any gasoline engine. This is primarily due to the higher compression ratio, requiring larger storage batteries when electrically started, as well as high-starting torque motors. In extremely cold weather, the difficulties are greatly increased.

6. Disadvantage of heavier reciprocating parts. The Diesel engine is not so flexible as the gasoline engine: on account of the heavier reciprocating parts it does not respond to the throttle as quickly as does the latter. There is likewise more vibration in the Diesel engine than in the gasoline as a result of these heavier parts coupled with the higher explosion pressures.

7. Lubricating difficulties. The Diesel operation presents more lubricating problems than does the gasoline. The filters require more frequent cleaning. The fuel oil must be efficiently filtered in order to remove all dust and foreign matter before the fuel is injected. The fuel injection pumps, spray nozzles, and valves must be protected from wear and scoring that result from dirty fuel. Dirty fuel can also cause sticking of close-fitted parts of the fuel injection apparatus.

8. Higher cost. It is obvious that for the Diesel there is a higher cost for service, maintenance, and parts. There are far too few skilled mechanics and maintenance men who are thoroughly familiar with the Diesel engine, and good Diesel experts are paid a much higher wage. This contributes importantly to the over-all cost of Diesel power for small and isolated installations.

Owners of Diesel engines have been willing to accept these annoyances and disadvantages for the sake of savings in fuel and operating costs, and for the other advantages of the Diesel

engine. This is very true in the heavy truck, tractor, bus, locomotive, and industrial fields, not to mention all kinds of marine applications.

Great progress has been made in the design of the Diesel engine since its invention by Rudolph Diesel about 1893. As a result of a half century of development, the relative power output has been greatly increased, the design and construction greatly simplified, and its cost reduced to a basis comparable with that of the gasoline engine.

Principle of operation. The high thermal efficiency or the low fuel consumption of the Diesel engine is ascribed to its principle of operation. As shown in Fig. 1-1, the piston compresses only pure air in the cylinder, to about 500 pounds per square inch (psi). The fuel is injected at near top dead center and is ignited by the heat generated by the compression of the air, around 1000° F. The resulting pressure of expansion forces the piston down, thus producing torque. Since the temperature of 1000° F is sufficient to ignite the fuel, no spark plug is needed for ignition of the air-fuel charge.

High compression ratio. The Diesel engine is more efficient than the gasoline engine owing to the fact that its compression ratio is higher. The compression ratio required depends upon the nature of the fuel. In the gasoline engine the compression ratio is limited to approximately 6 to 1. If the compression ratio is higher than practical for the fuel used, the fuel mixture will ignite before the compression has been completed. This causes detonation. The Diesel engine has a compression ratio of 15 to 1; this means that the air is compressed one fifteenth of its original volume. This also means a higher expansion ratio, and the more a gas expands after its combustion, the more power it produces, as explained by Fig. 1-2.[1] The net work an engine produces is the work of expansion minus the work of compression. Fig. 1-3 shows how the efficiency increases with compression ratio as well as with net work.

Air-fuel ratio. In order to ignite, the air-fuel mixture in the gasoline engine must not vary much from the theoretical or chemically correct air-fuel ratio of 14.5 pounds of air to 1 pound of fuel, a ratio that is obtained by throttling the intake charge through the carburetor. Throttling the intake charge increases the pumping losses and the frictional resistance to the flow of

[1] The graphs in Figs. 1-2 to 1-14 are based on calculations of a 5-in. bore by 6-in. stroke single-cylinder test engine at 800 rpm.

air through the carburetor, and, at the same time, reduces the compression pressure. The percentage of heat losses increases at part load, since in the gasoline engine the temperature remains about as high at part load as at full load.

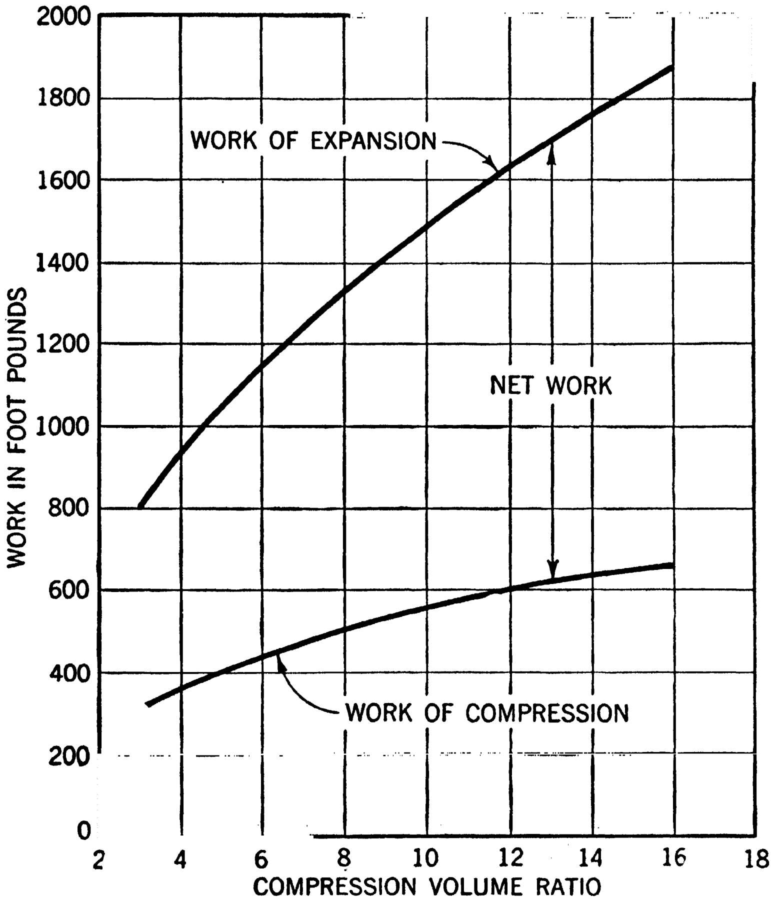


FIG. 1-2. Work of expansion increases at a faster rate than the work of compression as the compression ratio is increased. Based on a 5″ × 6″ single-cylinder test engine at 800 rpm.

The Diesel engine compresses a charge of pure air, and not an air-fuel mixture from a carburetor. This tells the reason for the Diesel's fuel economy at part load. With each stroke of the piston, the Diesel takes in a full charge of air, the amount of fuel injected being varied to suit the load on the engine. How

the Diesel air-fuel ratio varies all the way from approximately 20 to 1 at full load to as lean as 100 to 1 at idling, compared with the constant gasoline engine air-fuel ratio of 14.5 to 1, is shown in Fig. 1-4. This explains the chief difference between the

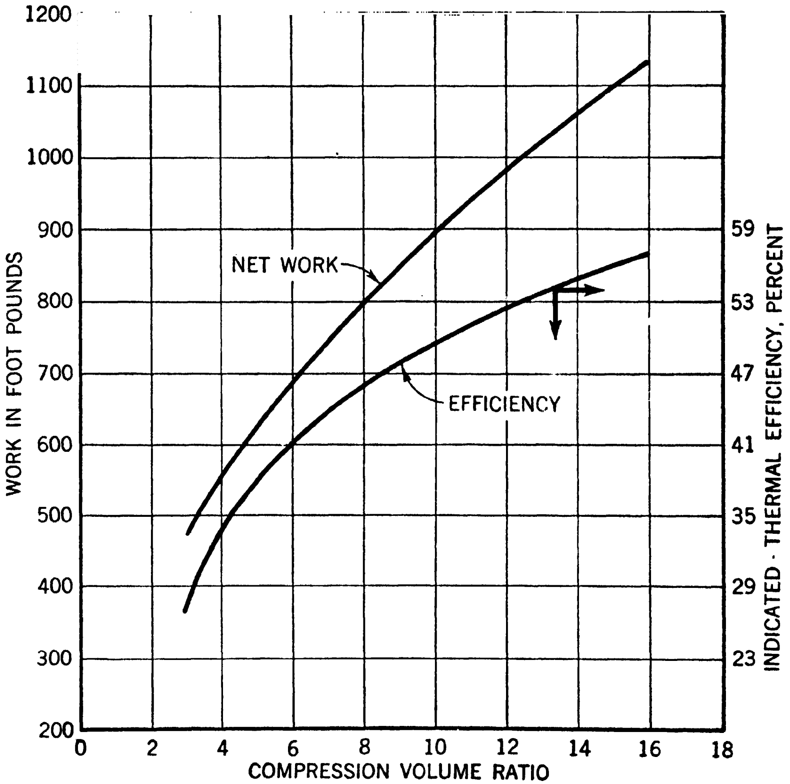


FIG. 1-3. Since net work increases with the compression ratio, efficiency also increases, but not at the same rate, because of increased friction at higher compression ratios.

gasoline engine and the Diesel engine, showing that the nature of the charge in the carburetor engine and that of the compression-ignition engine are two different things. The carburetor engine requires a homogeneous charge of fuel and air of about 14.5 to 1, with the fuel in vaporized form, regardless of the load on the engine. In the Diesel engine, the mixture ratio varies from a very lean to a very rich *stratified* charge. Instead of being homogeneous, it is heterogeneous. The fuel is injected

into the cylinder of the Diesel engine in the form of a spray. This fuel spray varies from a dense liquid core to a thin fog around it, varying in density from the center of the spray to its outer boundary. The tiny globules of fuel ignite, each inde-

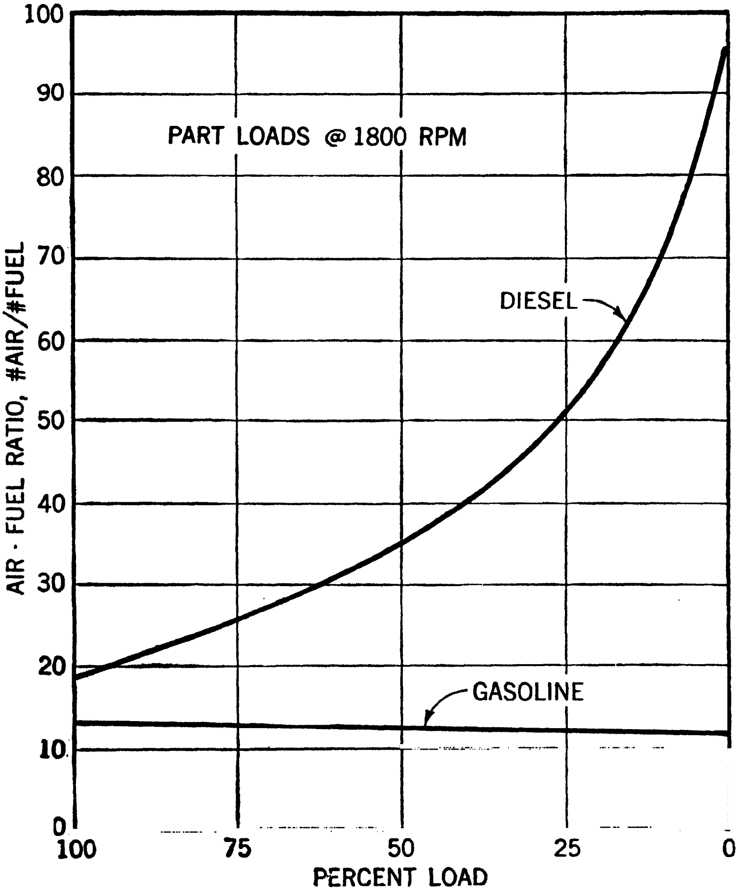


FIG. 1-4. Comparison of fuel-air ratios of Diesel and gasoline engines at various loads.

pendently, the combustion taking place at many different points in the combustion chamber at the same time, with the combustion spreading and the fuel combining with the air immediately around each point of initial combustion. An explosive mixture is not necessary in the Diesel engine, whereas it is essential in the gasoline engine.

Limitations of the Diesel cycle. It must be understood that not all the advantages are on the side of the Diesel; in fact, for certain installations, the Diesel engine may be decidedly inferior to the gasoline engine. There is a maximum horsepower that can be produced by a single unit practically and economically, and for very small engines, starting requirements must be considered. It may therefore be said that the Diesel's stratified charge with varying mixture ratio has its advantages

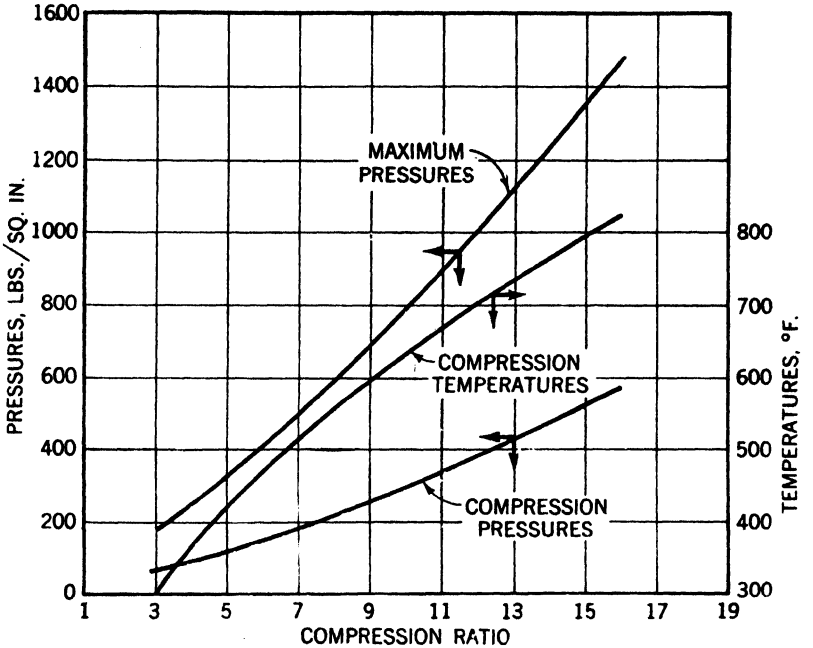


FIG. 1-5. Maximum pressures, compression pressures, and temperatures versus compression ratio.

and its disadvantages. And these must be understood by all who deal with Diesel power. While the Diesel has good fuel economy at part load and the charge need not be throttled, the fact is that it may have to waste air or fuel on account of the short injection period, during which the mixture of the fuel and air must take place. This particularly is a disadvantage at high speed and aggravates the maintenance problem. The approximate time for mixing the charge compared with speed or revolutions per minute in the Diesel engine and in the carburetor engine is shown in Fig. 1-10. At best, in the Diesel

all the fuel molecules cannot find all the oxygen molecules during the short time for the expansion stroke of the engine. Even at 1000 revolutions per minute (rpm)—and most high-speed engines run faster than that—the whole expansion stroke takes place in $\frac{1}{36}$ of a second. (For a 4-cycle engine, there are 500 expansion strokes divided by 3600 seconds, and further divided by the per cent of the cycle occupied by the expansion stroke.) (See Fig. 1-8.)

If the fuel is to burn efficiently, the mixing must be accomplished in the early part of the expansion stroke. Because the injection of the fuel and its mixing with the air cannot start much before the beginning of the expansion stroke, it must be accomplished in the shortest time possible so as to shorten the time for combustion as much as possible. How the thermal efficiency is directly related to combustion time in percentage of stroke is shown in Fig. 1-8. Therefore, we are dealing with a very short time interval, a condition that is extremely important in relation to the timing of the fuel injection of the Diesel engine.

Fuel-mixing problems. Even though the fuel spray is perfectly atomized and very evenly distributed, the fuel and air cannot mix completely in the short time available. To understand how this limitation is accepted and how the problems involved are contended with in the Diesel engine, it is necessary to consider the nature of combustion in this type of engine.

The chief problem of combustion in the Diesel engine can be simply explained. For the gasoline engine the carburetor does the mixing of the fuel and air in correct explosive mixture outside the cylinder. Therefore the gasoline vapor burns without difficulty, since it is a chemically correct homogeneous mixture. The Diesel must ignite and burn a stratified charge or heterogeneous mixture. Even if the air-fuel charge in the Diesel were without excess air, it would not be perfectly mixed. Some of the fuel particles fail to find the corresponding oxygen molecules in the short time available for the mechanical mixing of the fuel and air after injection. As a result, at the end of the expansion or power stroke, both unburned fuel and plenty of oxygen are found in the exhaust gases. To keep this unburned fuel to a minimum by correct timing and proper combustion is a maintenance and operative problem.

Chemistry of combustion. Combustion takes place when one carbon atom combines with two oxygen atoms, the combustion

reaction being usually expressed by the following equation:

$$C + O_2 = CO_2.$$

In the same way, two hydrogen atoms combine chemically with one of oxygen, with the reaction

$$2H_2 + O_2 = 2H_2O.$$

Now the fuel for the Diesel is similar to *heptane*, a hydrocarbon expressed by the formula C_7H_{16}, and, when complete or perfect

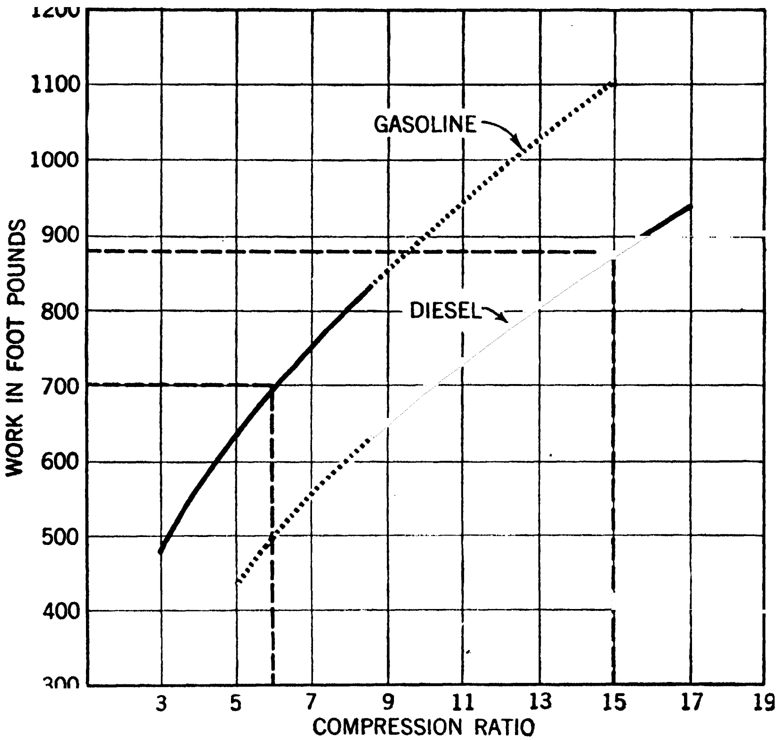


FIG. 1-6. More net work can be obtained from the Otto cycle than from the Diesel cycle at various compression ratios.

combustion takes place, this hydrocarbon combines with 11 oxygen atoms as follows:

$$C_7H_{16} + 11O_2 = 7CO_2 + 8H_2O.$$

The problem is that Diesel fuel is not any single chemical compound, but a mixture of hydrocarbons, whose components are various hydrocarbon molecules, but as far as combustion is

concerned the fuel ultimately consists of nothing but carbon and hydrogen. A study of Diesels indicates that the carbon-hydrogen ratio runs from 6 to 8 by weight depending on where the fuel came from and how it was refined.

Now 1 lb of hydrogen requires $^{32}\!/_{4}$, or 8, lb of oxygen, and 1 lb of carbon requires $^{32}\!/_{12}$, or 2.66, lb of oxygen for complete combustion. Since 1 lb of air contains 0.2315 lb of oxygen, 1 lb of hydrogen requires 8/0.2315, or 34.55, lb of air, and 1 lb of carbon requires 2.66/0.2315, or 11.519, lb of air for complete combustion. Hence it follows that 1 lb of Diesel fuel containing 1 weight of hydrogen to 7 weights of carbon needs for complete combustion:

$$\tfrac{1}{8} \text{ of } 34.55 + \tfrac{7}{8} \text{ of } 11.519 = 14.5 \text{ lb of air.}$$

This is the chemically correct, or theoretical, amount of air to burn 1 lb of fuel. Therefore, if the combustion chamber burns 1 lb of Diesel fuel and 14.5 lb of air, completely mixed, each particle of fuel must be surrounded with its required number of oxygen particles. There is complete combustion, with no fuel and air left over in the exhaust gases. In the gasoline engine this can occur, but in the Diesel, *no*. The fuel in the Diesel charge is at least one third less than would be permissible for a chemically or a theoretically correct mixture. Instead of 14.5 air-fuel ratio, the Diesel engine employs at least 50 per cent excess air, which means 1.5 × 14.5, or 21.8 lb of air. So, the Diesel wastes air to save fuel. Any attempt to use the chemically correct mixture would result in a certain portion of the fuel failing to find and combine chemically with the necessary air and hence would be found unburned in the exhaust gases. Fig. 1-11 aids in understanding this problem.

Excess air. It is evident that in order to mix the fuel and air in the Diesel engine in the extremely short time required, an excess of air considerably above the chemically correct ratio must be supplied. This may be anywhere from 25 to 90 per cent at full load, depending upon the design of the engine and the means of bringing about a rapid and thorough mixing of the fuel. If the design of the injection system and combustion chamber is such that the fuel and air mix quickly and completely, the excess air may be small. For some engines, in which the fuel globules cannot readily reach the far corners of the combustion chamber and any air pockets in the cylinder head, and hence cannot mix intimately with the air encountered,

considerable amount of excess air will be needed. The amount of excess air required must be carefully calculated and designed in order to reduce to a minimum the unburned fuel, or the engine fuel consumption will be high, the engine will smoke excessively, soot will be deposited, and the engine will be overheated, thus creating difficult maintenance problems.

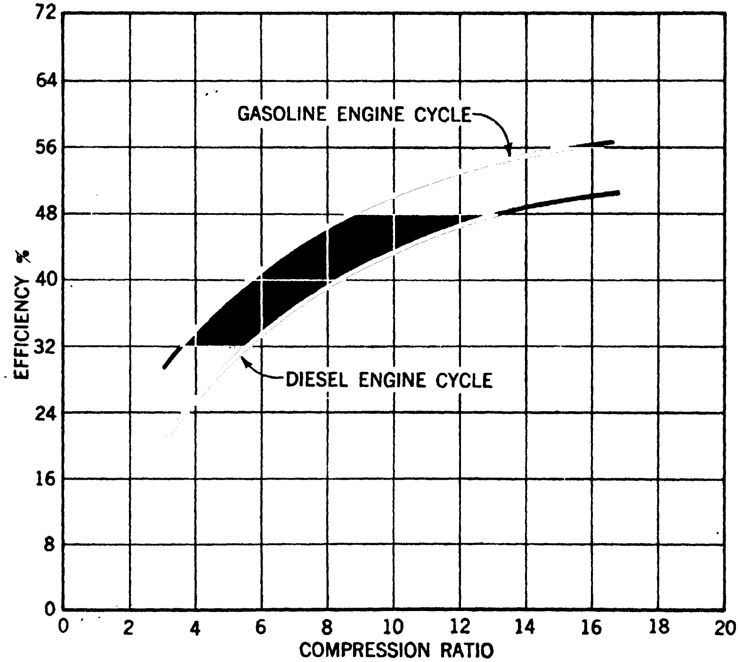


FIG. 1-7. The theoretical thermal efficiency of the gasoline engine is higher than that of the Diesel engine at all compression ratios. As shown in Fig. 1-6, the Diesel is able to operate at higher compression ratios than are possible with the gasoline engine.

Effect of excess air. It is unfortunate for the Diesel engine that the need for this excess air reduces the power output that can be obtained from a Diesel cylinder of a given size. Fig. 1-6 shows that for the same compression ratio more net work can be obtained from a gasoline engine cylinder than from a Diesel engine cylinder of the same size. Hence, the thermal efficiency of the gasoline engine is higher than that of the Diesel engine at the same compression ratio. If the gasoline engine could be operated at the same high compression ratios as the Diesel,

there would be no need of a Diesel engine. It is reasonable to assume that continued efforts will be made to design gasoline engines to operate at higher compression ratios. The fact is that the gasoline engine has a higher-thermal efficiency because it burns more nearly the theoretically correct air-fuel mixture than does the Diesel, as shown in Fig. 1-7. There is, however, another scientific reason, equally important for the higher efficiency of the Otto cycle, as indicated in Fig. 1-12. The gasoline or Otto cycle engine burns a greater amount of the fuel at *constant volume*, while the Diesel burns a greater amount at *constant pressure*. This can be completely explained only by thermodynamics, but it is reasonably comprehensible. Since the Diesel engine usually cannot be operated satisfactorily with less than 50 per cent excess air, it follows that one-third less fuel is burned in a Diesel cylinder than in the gasoline engine cylinder of the same size and revolutions per minute. Since Diesel fuel and gasoline have roughly about the same heat energy per pound, around 19,000 British thermal units (Btu), it naturally follows that the power output of the Diesel cylinder will be one-third less than that of the Otto cycle engine of the same size, speed, general principle of operation, and design. Then the question arises: *Why is a Diesel engine?*

Efficiency of the Diesel engine. The efficiency of the Diesel is higher than that of gasoline engine, simply because the gasoline engine must operate at a lower compression ratio. Detonation and preignition take place at any higher compression ratio. The Diesel engine operates at relatively high compression ratios, and the higher the compression ratio the better the efficiency, as clearly shown in Fig. 1-7, derived from Fig. 1-6.

Another reason why the Diesel engine has greater efficiency than has the gasoline engine is the lower specific fuel consumption of the former. Although not so much fuel can be burned in the same-sized cylinder for the Diesel engine, 4 lb of Diesel fuel produces as much power as 5 lb of gasoline. This reduces the power-output handicap just shown for the Diesel engine to $5/4 \times 2/3$, or $10/12$. Taking the gasoline engine power output at 100 per cent, the output of the Diesel is 84 per cent instead of $66\frac{2}{3}$ per cent, provided we consider the same size, speed, and so on, of the two engines.

Fuel injection difficulties. Since excess air needed in the Diesel engine was the chief reason for the lower Diesel power output, it was logical that considerable experimentation and

research were done during the entire history of the Diesel engine to lower this margin of excess air. Many difficulties beset the designers working on this problem, and, while much success has been achieved, difficulties of operation and maintenance were encountered, particularly in the development of high-speed engines for uses competitive with the gasoline engine.

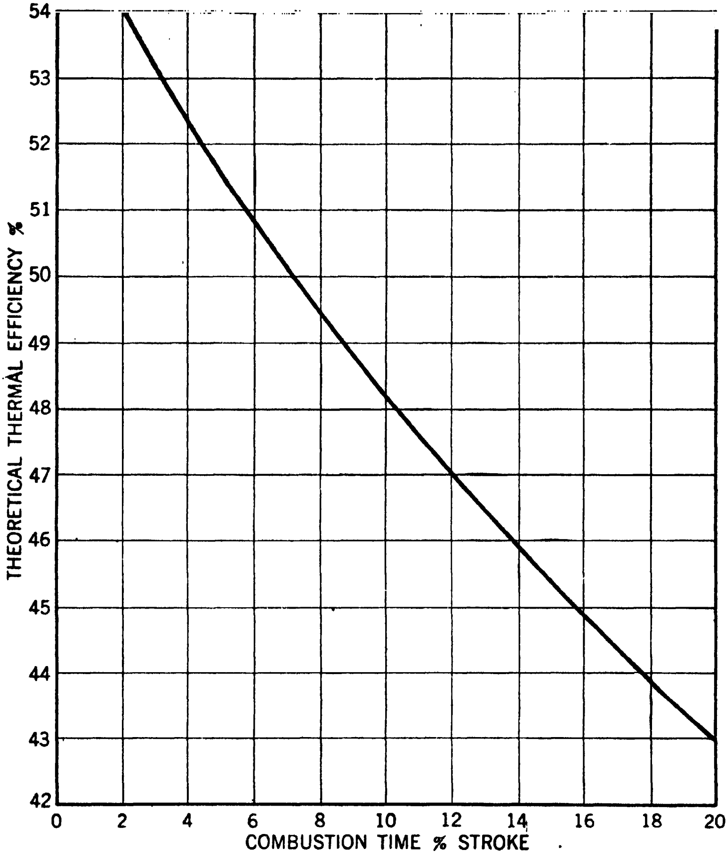


FIG. 1-8. Relation of efficiency to combustion time in per cent of stroke.

The early Diesel engine, heavy-duty, air injection, and slow-speed type, reached a high state of development as long as twenty-five years ago. The air injection provided an excellent means of mixing the fuel and the air, and the excess air required for such engines was relatively low. Moreover, the slow speed provided considerable time for mixing and combustion.

When the elimination of the air injection compressor was attempted and solid injection introduced to take its place, along with the design of higher-speed engines, new difficulties beset the path of the designer. Fuel injection pump and nozzle troubles were encountered and fuel-mixing problems showed up; high-pressure injection required small, high-precision fuel injection pumps. The problems of fuel atomization, dispersion, and penetration were aggravated, and many years of toil and sweat were needed to make progress in their solution. The fuel had to be broken into fine droplets, or the spray failed to mix intimately with the air, and much smoky exhaust resulted that could be cleared only by introduction of excess air in larger quantity than formerly required for the air injection engine. Then the output per cubic foot of piston displacement was found to be less than desired. Since smoky exhaust could not be tolerated in submarines and providing excess air required a larger engine than formerly used, our Navy actually did not consider solid injection for submarine Diesels until about ten years before World War II, when development of solid injection engines had finally reached a satisfactory stage.

Progress in design. While complete success has by no means been realized by the Diesel designer, gradual improvements produced modern injectors working to tolerances closer than found in any other mechanical device. It is now possible to produce finely atomized sprays and to distribute them accurately and uniformly throughout the air spaces in the combustion chambers. Designers have shaped the combustion chambers to fit the form of the spray, and so improved the mixing efficiency, thus reducing the excess air margin to a very great extent. Many operating troubles have also been eliminated, but enough still remain to require the best efforts of the maintenance man and the operator.

1. Improvements made to solve problems. The Diesel spray must be able to penetrate a dense 500 psi air, and this becomes very difficult when the nature of the spray is considered. A finely atomized spray can be produced and well dispersed in one atmosphere, but to produce it in 15 atmospheres of pressure is quite another thing. In the first place, the spray must be divided by means of multiorifice nozzles into several sprays in order to reach the entire combustion space. The spray globules must be broken down to tiny droplets $1/1000$ in. in diameter, and this requires injection at several thousand pounds per square

inch. Fine droplets are essential to rapid ignition and these must be injected at a velocity that is very great, sometimes as high as 400 mi a second. To attain such a velocity, extremely small spray holes in the nozzle must be provided. And with small holes and fine droplets, the resistance of the air makes

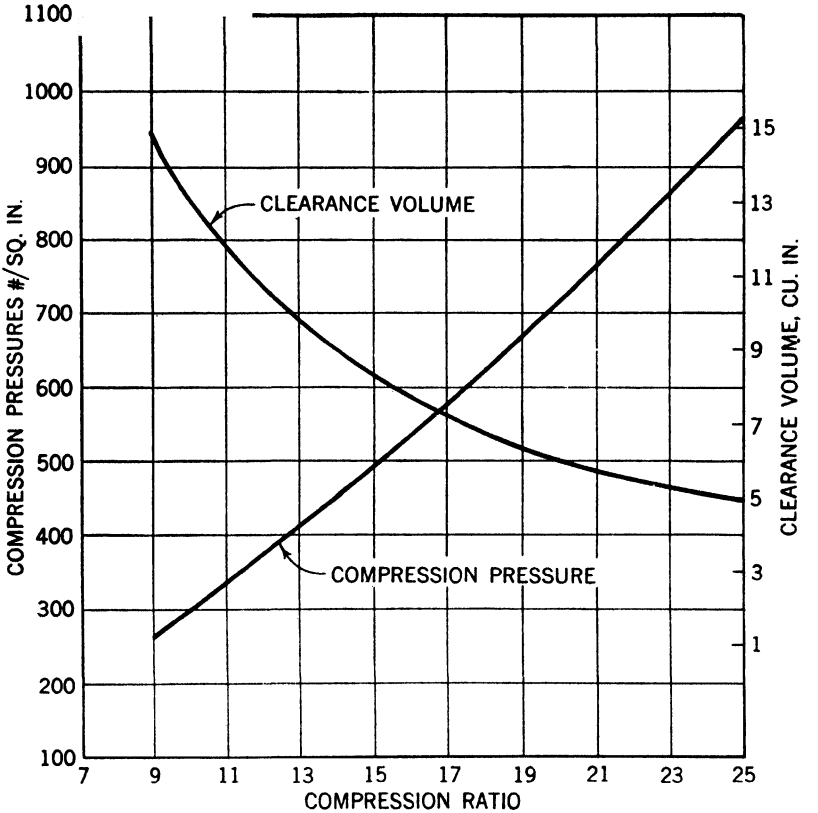


FIG. 1-9. Relation of compression pressure to clearance volume. In extremely small combustion chambers, the fuel strikes the walls and does not burn effectively, because it cools and does not mix readily with the air.

penetration very difficult. The same spray which would be projected yards in one atmosphere could travel only inches in the dense air compressed in the combustion chamber. Many of the droplets fall short, their momentum being rapidly neutralized by the density of the air. A point is reached where the size of the droplet and the injection pressure make the penetration and dispersal less because the size of the globules is

so small that momentum is soon overcome. It is thus that fine atomization and deep penetration work against each other, and tend to neutralize each other, yet both are required. The designer must get the maximum of each without sacrificing one or the other.

2. The use of air turbulence. When it was no longer possible to get the fuel to the air, methods of bringing the air to the fuel spray were attempted, and with considerable success.

A number of methods have been tried and an effort made to set the combustion air in motion and whirl it around after it entered the cylinder in order to mix it better with the fuel spray. It was found that admitting the air tangentially induced it to perform a rotary motion in the cylinder. Several engines have a separate swirl chamber into which the air is forced, where it rotates while the fuel is being injected. Other engines are designed with precombustion chambers, in which the initial stage of the explosion takes place. This preliminary explosion greatly agitates the air and fuel, and, as the mixture is blown out into the main combustion space, further mixing takes place. The air-cell type of chamber was developed to exploit further this possibility, and, owing to the preliminary explosion that issues from it, great turbulence and agitation of the combustion gases occur during the time the combustion is taking place. A number of devices, now more or less successfully used in American Diesel engines, having for their purpose a better mixing of the fuel and air in the shortest possible time, have been developed. These consist of the precombustion chamber, the energy cell, and the air cell engines, each of which has a great deal of merit.

3. Effect of new designs. Some disadvantages are introduced as a result of the special types of combustion chambers. Excess air could not be dispensed with to any considerable extent. When chambers were designed that reduced the excess air to a value as low as 25 per cent, fuel economy was sacrificed as fuel consumption increased on account of the higher mechanical and heat losses the use of the design entailed. High velocity of the air was necessary to obtain high turbulence for mixing the fuel with air, and high velocity of the air required more of the engine's power. Moreover, turbulent air gives more heat up to the cylinder walls than does ambient air. It is said that the precombustion chamber engines average more than 10 per cent higher fuel consumption than do the plain open combustion

chamber, and the power output of the former is not increased over that of the latter.

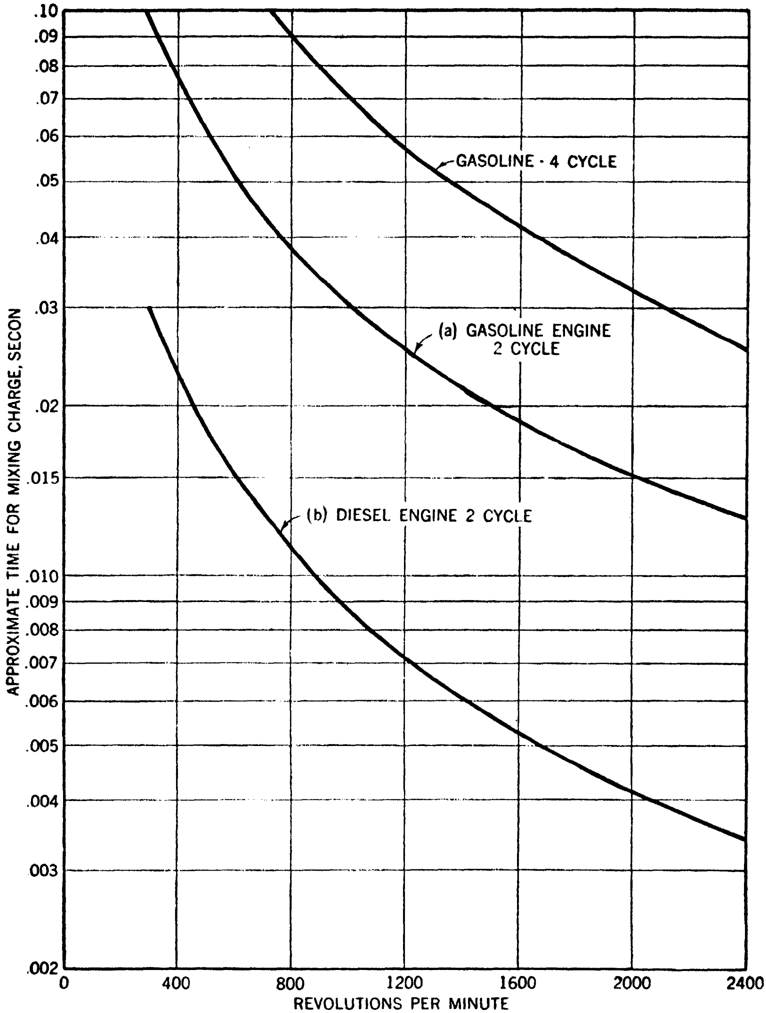


FIG. 1-10. The mixing time in high rpm engines is so short that the advanced injection timing necessary creates high maximum pressures, which impose severe stress on the engine. Maximum pressures may exceed 1200 lb.

There were distinct advantages, however. Such engines had more flexibility and avoided the use of high-pressure sprays and small orifices. A coarse spray and a single-hole orifice nozzle with comparatively low injection pressures gave satisfactory

performance when sufficient air turbulence was obtained to mix the fuel and air thoroughly.

Methods of increasing power output. Early Diesel engines ran at operative speeds of 125 to 250 revolutions per minute. As the need for greater output for the engine size came to be recognized rotative speed was gradually increased to 500, 600, and 1000 rpm. Only by means of improved lubrication, lighter reciprocating parts, and the introduction of solid injection did the Diesel engine have any chance to become a high-speed competitor of the gasoline engine. For fifteen years, the development of the high-speed engine has progressed rapidly throughout the world. At the present time, tractors, buses, and trucks are equipped with Diesel engines. These engines run twice as many miles on a gallon of fuel as the carburetor engines they replace.

The maximum rotative speed of present Diesels for automotive applications is more than 2000 rpm. This has been made possible by the high-precision fuel injection pump, so well designed and built that it meters fuel quantities as small as a drop of water, under extremely high pressure, and during a time interval measured in a thousandth of a second.

Difficulties of increasing power output. Although there are reasons to believe that continued progress in this direction is to be expected, two obvious difficulties stand in the way of additional power boosts.

1. Problem of mixing air and fuel. The excess air and the mixing process have already been discussed. Good combustion requires that the air and fuel mix in the early part of the power stroke; this is to avoid delay in completion of ignition and combustion. By referring to Fig. 1-10 it should be evident that mixing the fuel and air in a short time constitutes a difficult problem. At 2400 rpm with a crank angle of 30 deg for fuel injection, there is only $\frac{1}{480}$ of a second in which to mix the fuel and air intimately and burn the mixture.

2. Ignition lag. The second most difficult problem is that of ignition lag or a delay in the start of combustion. The fuel does not ignite the instant it encounters the highly heated air. There is a definite, measurable delay, said to be around 0.002 sec. While this seems a short interval, it corresponds to 28.8 deg of the crank rotation; in other words, the crank passes through that space during 0.002 sec. If the engine timing is not advanced to allow for this delay, ignition may fail to take place and this failure may cause the engine to misfire. When ignition

does not start promptly, the fuel burns faster and more violently when it does ignite. This causes combustion knock. The fuel accumulates in the cylinder during the ignition delay period. The duration of the injection period corresponds to 15 to 30 deg of rotation, depending on the design of the engine. Now if the greater part of the fuel charge is already in the engine when ignition starts, the entire quantity will ignite at practically the

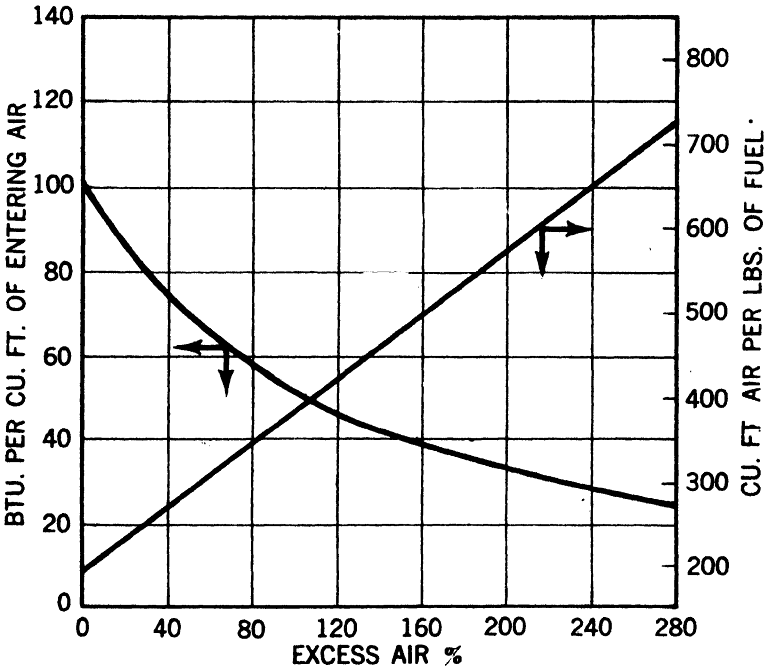


FIG. 1-11. High excess air reduces heating value per cubic foot of air-fuel mixture, which reduces the output of the engine.

same time; the explosion is violent, and the knock and vibration may be disastrous to the engine. Diesel combustion knock is similar to detonation in the gasoline engine. There is a definite difference between the two, however, which makes it necessary to understand this behavior of the Diesel engine.

The gasoline engine knocks because of preignition, ignition *too early*, while the Diesel knocks because of a *delay* in the start of ignition. However, the fundamental cause of the knock is the same, namely, fast burning and consequent excessive rate of pressure rise. An engine runs smoothly if the maximum

pressure rise does not exceed 30 psi per degree crank rotation. If the rate of pressure rise goes above 50 psi per degree, the engine most certainly knocks and imposes heavy strain on the bearings and other reciprocating parts and reduces the power output accordingly, as shown in Fig. 1-13. This curve of maximum cylinder pressures shows that no further increase in power results when the maximum pressure exceeds 1200 psi, owing to the excessive friction load. Bearing and cylinder maintenance problems increase accordingly.

Diesel combustion knock. Rapid pressure rise takes place in the gasoline engine, and the knock comes at the last stage of combustion. It is the last portion of the fuel to burn that causes the knock. Only a small part of the gas adjacent to the spark plug ignites at first. As propagation of the combustion gradually proceeds, the flame front expands and in doing so compresses the unburned part of the cylinder contents. Finally, the unburned part is compressed by the rapid pressure to the self-ignition point. Then the entire charge is ignited, producing a rapid pressure rise, heard as sharp knock or detonation.

In the Diesel engine, the occurrence is quite different. Combustion begins with spontaneous ignition of the fuel, but this does not cause the ignition to knock; the determining factor is how much fuel is present and ignites, in the entire mass, at the same time. The greater the quantity to ignite at one time, the greater is the pressure rise and the more severe the knock. In this case, the ignition lag gets in its work. The fuel fails to ignite instantaneously, for it requires a delay period for the fuel to start burning and this delay period depends upon the quality of the fuel, the cylinder pressure, the temperature of the air, and other factors. It is evident that the longer the lag, the greater the amount of fuel that accumulates in the cylinder before ignition actually starts. When this does occur, nearly all of the fuel now piling up in the cylinder ignites simultaneously. Consequently, the longer the delay period or ignition lag, the more severe will be the knock.

The ignition characteristics of Diesel fuels vary over a rather wide range. The qualities that reduce the ignition lag suppress the knock in the Diesel engine. High compression ratio, warm intake air temperature, and high pressure help speed up ignition, shorten the ignition lag, and suppress the tendency to knock.

Cetane. In the gasoline engine, high *octane* fuels burn more slowly than other fuels and thus prevent detonation in the

engine. This situation is the opposite of what is desired in the Diesel engine. What are known as high *cetane* fuels are used in the Diesel engine. They ignite without too much ignition lag. Diesel fuel specifications include the cetane number of the fuel. Some high-quality fuels may be nearly 100

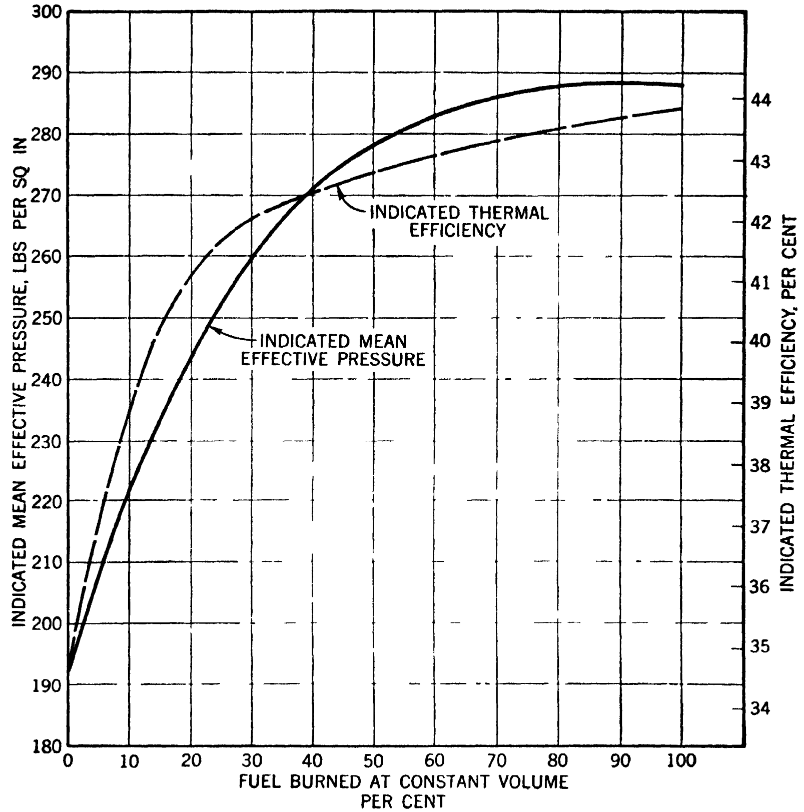


FIG. 1-12 High efficiency is related to amount of fuel burned at constant volume hence the higher theoretical thermal efficiency of the Otto cycle engine

cetane, which means that they have extremely short ignition lag—as low as $^1/_{1000}$ sec. This is about the limit; further improvement is not expected. This fact constitutes a serious limitation on increasing the speed of the Diesel.

Two- and 4-stroke cycle. Increasing the speed, however, is not the only means of getting more power out of the Diesel engine. Other means of reducing the weight per horsepower

output have been more successful. Although the factors that control the power output of an engine are thermal efficiency, excess air, high rotative speed, and the amount of air that can be charged into the cylinder per revolution, it is more power strokes per minute that have shown the greatest results. For

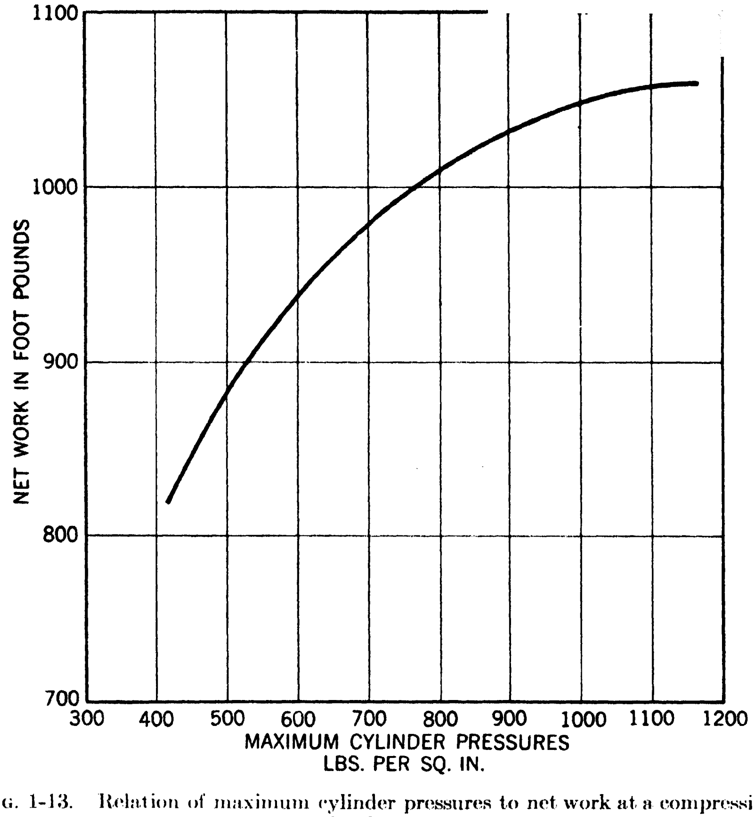


FIG. 1-13. Relation of maximum cylinder pressures to net work at a compression ratio of 13 to 1.

the 4-cycle engine, every other revolution is wasted so far as feeding air into the engine is concerned.

In the 4-stroke cycle engine, the piston sucks a charge into the cylinder, while in the 2-stroke cycle, there is no suction stroke. The charge must be forced into the cylinder by a pump or blower. In a 2-stroke engine both the exhaust and intake occur in the short time when the piston is at or near the bottom dead center. Thus, these two processes overlap, the fresh air

charge aiding in scavenging the cylinder of the burned gases. In a Diesel engine scavenging can be done with air without wasting fuel as is the case with the 2-cycle gasoline engine. Hence, the 2-stroke cycle design for Diesel becomes a very practical thing, although it is not considered for the usual

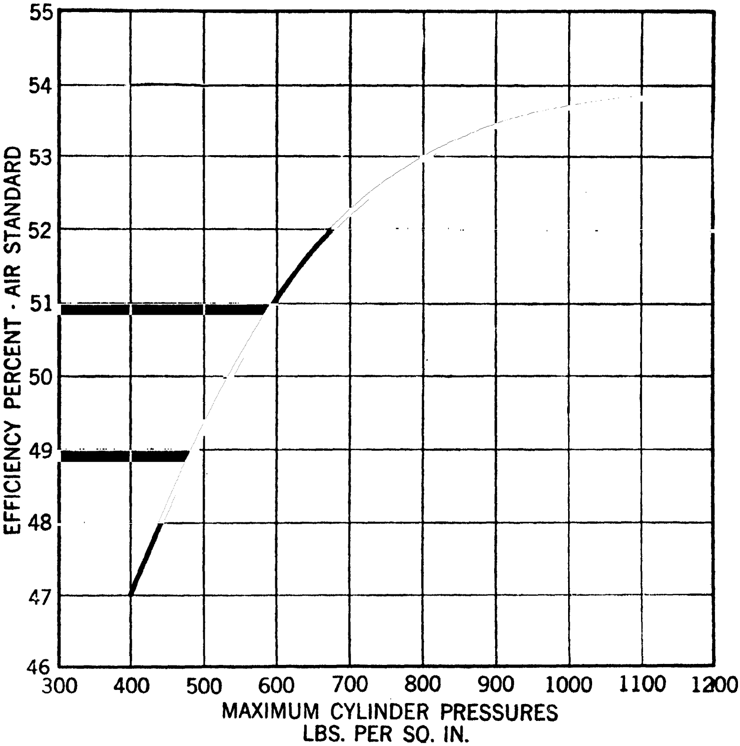


FIG. 1-14. Variation in efficiency with maximum cylinder pressures. With Fig. 1-13, it is shown that material reduction in maximum pressures without much reduction in useful work and efficiency is possible.

gasoline engine. The 2-stroke cycle Diesel has made significant advances in recent years.

The radial or "pancake" Diesel. One of the most advanced marine Diesel engines is the "pancake" engine, a radial with four banks of cylinders, one above the other. The engine's weight is around 4 lb per horsepower. It has a 6-in. stroke and a 6½-in. bore and is rated at 1200 hp at 1800 rpm. This 2-cycle engine has aircraft cylinder construction and is a

superior design with water jackets of corrugated sheet metal welded on the cylinder. A centrifugal blower operating at high speed furnishes the scavenging air with ports for intake and poppet valves for exhaust.

The opposed piston engine. Great importance attaches to the development of the opposed piston engine, a type of 2-

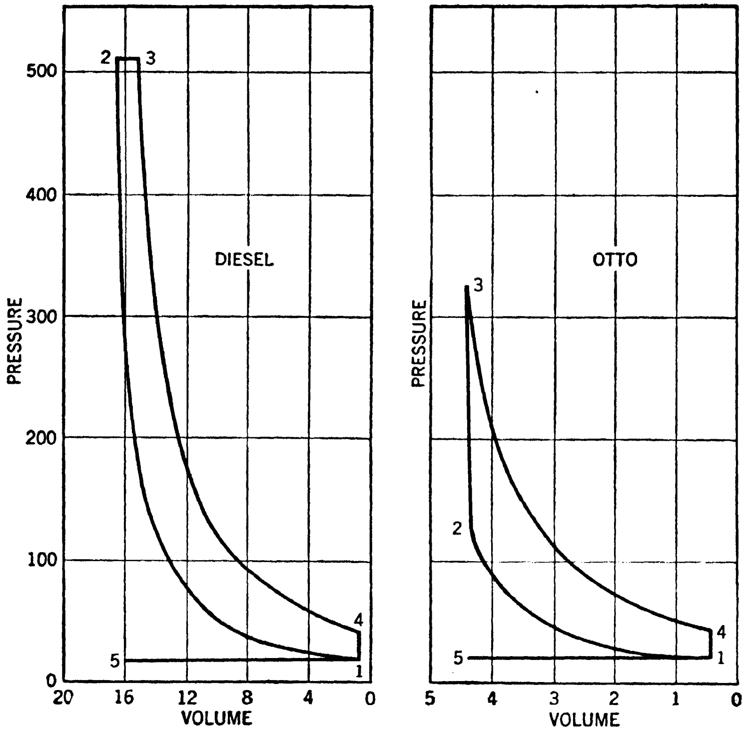


FIG. 1-15. Diesel and Otto cycle theoretical indicator diagrams drawn to same scale.

stroke cycle engine. A special advantage of this design is that it has no cylinder head, the explosion taking place between the opposed pistons. The loss that usually occurs through a cylinder head is therefore eliminated. Uniflow scavenging is another characteristic of the opposed piston engine, the air being admitted through a row of ports at one end of the cylinder and exhausting through a row of ports at the other end. One piston controls the inlet and the other piston controls the exhaust. The cylinder is efficiently scavenged and supercharged.

One of these engines has an 8-in. bore and a 10-in. stroke by each piston and operates at 720 rpm. There are two crankshafts, one on the bottom and one on the top, geared together so that the exhaust piston has a 12-deg advance over the intake piston. This lead permits the cylinder pressure to drop before the inlet ports open and effect some supercharging. These engines have very high fuel economy.

Another recent development is the double-acting engine, which is also 2-cycle. The opposed piston, the double-acting, and the 4- and 2-stroke cycle engines will be described fully in another chapter.

Uniflow or straight-through scavenging. Uniflow scavenging of the 2-cycle engine becomes a practical fact, and new and advanced Diesels of this design have appeared in recent years. The operation consists of intaking the air through ports in the cylinder near the bottom of the stroke and exhausting through ordinary poppet valves in the cylinder head. The principle of uniflow scavenging is more efficient than the so-called loop or cross scavenging, in which the air has to take several turns in going through a cylinder. The air for such engines is supplied by a blower at a pressure of around 5 psi, the capacity of the blower being about 40 per cent more than is necessary to fill the cylinder at atmospheric pressure. This extra air is employed in scavenging and supercharging.

The principle of uniflow or straight-through scavenging. For 2-stroke cycle engines uniflow scavenging is the most efficient method devised. This is due to the fact that around 20 per cent more air can be trapped by this means than by the cross or loop scavenging. Three methods have been devised to produce uniflow scavenging: (1) by poppet valves in the cylinder head, (2) by means of the opposed piston, and (3) by the use of the sleeve or slide valves. The sleeve-valve engine is a new design, in which the scavenging poppet valve is absent, leaving the cylinder head a simple symmetrical shape. While the scavenging is efficient, there are disadvantages, such as difficulty of cooling and lubricating the engine because of the sleeve. An aircraft engine of this type developing around 2000 hp on one shaft and weighing but 1.8 lb per horsepower has been built.

Supercharging. Increasing the air charge by supercharging has become a rather general practice for 4-stroke cycle Diesel engines. The volumetric efficiency is increased and the amount of the air charge exceeds the piston displacement at atmospheric

pressure. The result is more power per cubic inch of piston displacement. The chief purpose of supercharging the Diesel engine is to increase the power output. Since increasing the compression pressure does not cause combustion knock in the Diesel engines as it does in the gasoline engine, the use of this device on the Diesel engine has progressed rapidly. Supercharging and increasing the compression pressure actually make the Diesel engine run smoother. The supercharged 4-cycle Diesel engine is now a proved success, and most of the manufacturers of this type of engine provide supercharging.

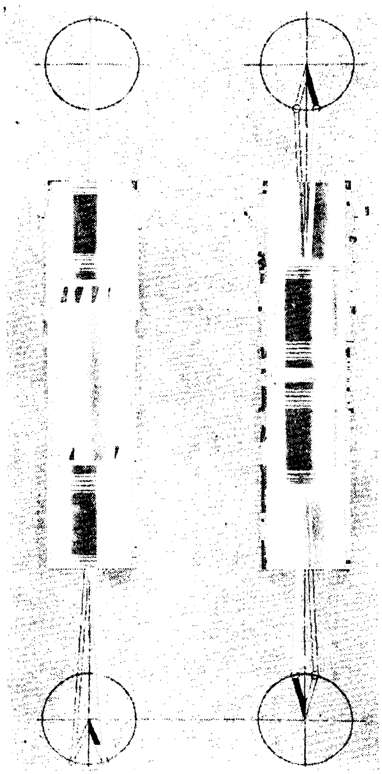


FIG. 1-16. Fairbanks Morse opposed piston engine.

There are two methods of supercharging—namely, the mechanically driven blower and the turboblower. There are several types of the mechanical supercharger: the reciprocating pump type, the rotary type, and the centrifugal blower, driven by the engine or an electric motor. The use of the supercharger permits the weight of a given engine to be reduced. A typical weight reduction is that of an engine of 150 hp, 6-cycle Diesel. This engine weighed about 14.5 lb per horsepower without supercharging. When the supercharger was added, the engine developed 200 hp and the weight per horsepower was 12.5 lb. However, there is a limit to supercharging that is economical. At the present time, this is found to be about 30 per cent increase in the power output. The power absorbed by the blower when this limit is exceeded becomes rapidly more until there is no further return from the power expended. The supercharger pressures range from 3 to 5 lb for average designs.

The turbosupercharger. The turbosupercharger is an exhaust gas turbine, invented by the French engineer Rateau, and perfected in this country by Dr. Moss of the General Electric Company. It is used extensively in aircraft for high-altitude

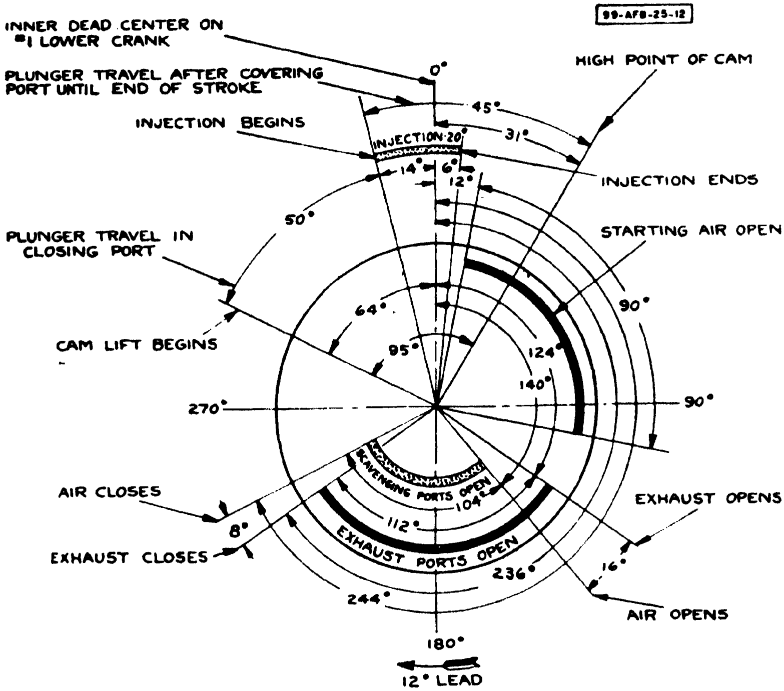


FIG. 1-17. Sequence and timing of events in a complete two-stroke cycle when the engine is running at full load and full speed. It can be seen that when the upper piston reaches its inner dead center in the compression stroke, the lower piston has completed 12° (the total crank lead) of its power stroke. This causes the lower piston to receive, at full engine load, the greater part of the expansion work with the result that about 72 per cent of the total power is delivered by the lower crankshaft. The remaining power is delivered to the upper crankshaft, where it is partially absorbed in driving the scavenging blower. This leaves only a relatively small amount of power to be transmitted through the vertical gear drive to the lower crankshaft, which is connected to the final drive.

supercharging. This method was adapted to the Diesel engine some years ago by the Swiss engineer Buchi, and was introduced into use in this country just before World War II. With the turbocharger, the power output was increased up to 40 per cent without sacrificing fuel economy.

The turbine of the Buchi supercharger is driven by the exhaust gas that comes from the engine at high pressure and temperature. A feature of this turbine is that the exhaust impulses are timed in such a manner that a major part of the energy of the exhaust gases is imparted to the turbine wheel without creating appreciable back pressure in the engine. High back pressure would seriously reduce the output and impair the engine's performance. Since the temperatures of the exhaust gases are as high as 1000° F, the turbine must be made to withstand this high heat. The turbine drives the centrifugal blower on the same shaft and blows air into the engine at 3 to 6 psi.

A practical limit on supercharging the Diesel engine is set by the process of engine cooling. The engine must reject as much heat through cooling as is developed into power. The more power the engines develop, the more heat that has to be removed by cooling from a given engine. It is for this reason that in highly supercharged Diesel engines piston rings stick, pistons are burned, and the exhaust valves and cylinder heads give trouble. Solution of these problems is regarded as a matter of time, when piston cooling, metallurgical improvements, better water-cooling systems, and improved methods of lubricating have been worked out.

QUESTIONS

1. Name three ways to increase the output of a given displacement Diesel engine.

2. What occurs when an attempt is made to burn fuel in a Diesel engine with a small amount of excess air?

3. What is the main reason why the Diesel engine is more efficient than the spark ignition engine?

4. The Diesel engine is relatively heavier than the gasoline engine. Why?

5. What are the advantages of the Diesel engine compared with the gasoline engine?

6. Under certain conditions, excess air in the Diesel engine can be reduced by means of better mixing of the fuel and air. What is necessary to do this?

7. Higher engine speeds for Diesel engines are limited by what?

8. The 2-stroke cycle Diesel engine is used in preference to the 4-stroke for certain applications. What is the main reason?

9. High-output 2-stroke cycle Diesel engines most commonly make use of what kind of scavenging?

10. The limits on supercharging a Diesel engine are what?

11. What type of Diesel engine is always simpler than others in construction?

12. Does better fuel spray design make possible the use of less excess air?

13. What limits the output of a Diesel engine: means and methods of cooling the piston and valves, or means and methods of supplying the engine with air?

14. What means are used to increase the time available for mixing the fuel and the air?

15. In the larger engine, what general class of combustion chamber is used predominantly, and why?

16. For small high-speed Diesel engines, what kind of combustion chambers are in common use, and why?

17. What is the purpose of air turbulence?

18. What kind of spray nozzles are used with the open combustion chambers, and what kind are generally used with the precombustion chambers?

19. What type of combustion chamber would be used for: (a) power output and greater mean effective pressure; (b) fuel economy; (c) ease of starting; (d) speed flexibility; (e) smoothness of operation?

20. Why is the divided, or precombustion, chamber harder to start?

21. What is one of the chief reasons for the higher cost of Diesels as compared with gasoline engines?

22. Why is the Diesel noisier than the gasoline engine?

23. What is the advantage of higher compression ratio with respect to efficiency?

24. How is the net work of the cycle determined?

25. What is an explosive mixture, and what kind of engine employs such a mixture?

26. Fuel is injected in the form of a spray in the Diesel engine. Is it atomized, vaporized, or mixed by turbulence?

27. If Diesel fuel was completely vaporized and mixed with the air, as in the gasoline engine, what would occur upon ignition of the charge?

28. What is the difference between a homogeneous and a heterogeneous mixture?

29. Is it necessary or not necessary to throttle the air charge in the Diesel engine? Why?

30. Why is it necessary to throttle the charge in the gasoline engine?

31. How and why is thermal efficiency directly related to combustion time in percentage of stroke?

32. How many pounds of air are required, chemically, to burn one pound of fuel in any engine?

33. Why is it necessary to supply more air than the engine requires or consumes in burning the fuel in the Diesel engine? Is this also true of the gasoline engine?

34. If the Diesel engine takes in 29 lb of air for each pound of fuel, what percentage of excess air is employed?

35. Theoretically, can more net work be obtained from the gasoline engine than from the Diesel engine, per cubic inch piston displacement?

36. If the gasoline could operate at 15 to 1 compression ratio, would there be a need for the Diesel engine?

37. Which engine may be said to waste air?

38. When a Diesel engine is operating with 50 per cent excess air, why is it that it can produce only two thirds as much power as the same displacement in the gasoline engine with practically no excess air?

39. What is meant by "lower specific fuel consumption"?

40. Why is the Diesel able to operate with a lower specific fuel consumption than is the gasoline engine?

41. Why must the gasoline engine operate with a lower compression ratio than the Diesel engine?

42. What problems were encountered by the designer when he undertook to reduce the amount of the excess air required?

43. When the designer eliminated air injection, and commenced the design of mechanical injection, what new problems were encountered?

44. Can a mechanical injection Diesel operate with as little excess air as the air injection engine? Why?

45. Since fine atomization and deep penetration work against each other, how does the designer get the maximum of each in order to accomplish as complete mixing of the air and fuel as possible?

46. What is the function of air turbulence?

47. What problems of maintenance are involved in the fuel injection process and system?

48. Why is a turbulence chamber or precombustion chamber engine less efficient than the plain, open-type combustion chamber?

49. What is ignition delay, and what is the reason for this lag?

50. Why does a Diesel engine knock, and how does this knock differ from that of a gasoline engine? How does this knock, in either engine, affect the maximum pressures?

CHAPTER 2

PROBLEMS OF APPLICATION

THE fundamental problems inherent in the operating principle of the Diesel engine were outlined and discussed in the preceding chapter, in which the limitations of the Diesel engine were illustrated by graphs. The phenomenon of combustion by the heat of compression, the mechanics of fuel injection, and spray atomization were detailed to show how the fundamental problems were related to maintenance and operation. The development and application of the Diesel involved problems of maintenance and operation. It is now in order to give the reader the proper perspective and a sound viewpoint on the technical problems encountered in the design and construction of the Diesel engine, and the solutions worked out by practical experience that may be brought to bear upon the related problems of maintenance and operation.

Immediately following World War I, intensive development of Diesel engine design occurred in Germany, England, and other European countries—especially in Germany. In the United States any marked progress was difficult to discern until about 1930, but since that time great strides have been made. Many improvements in engine design in Europe were later incorporated in American designs. Since Diesel engine interest was worldwide, successful European designers licensed some American manufacturers to build engines after their tested models. Although in Europe the automotive Diesel and engines for locomotives and submarines outdistanced other applications, such as buses, trucks, and tractors, Diesel engines in the latter applications greatly increased, both in this country and abroad.

Numerous problems of operation and maintenance were encountered that induced a great number and variety of

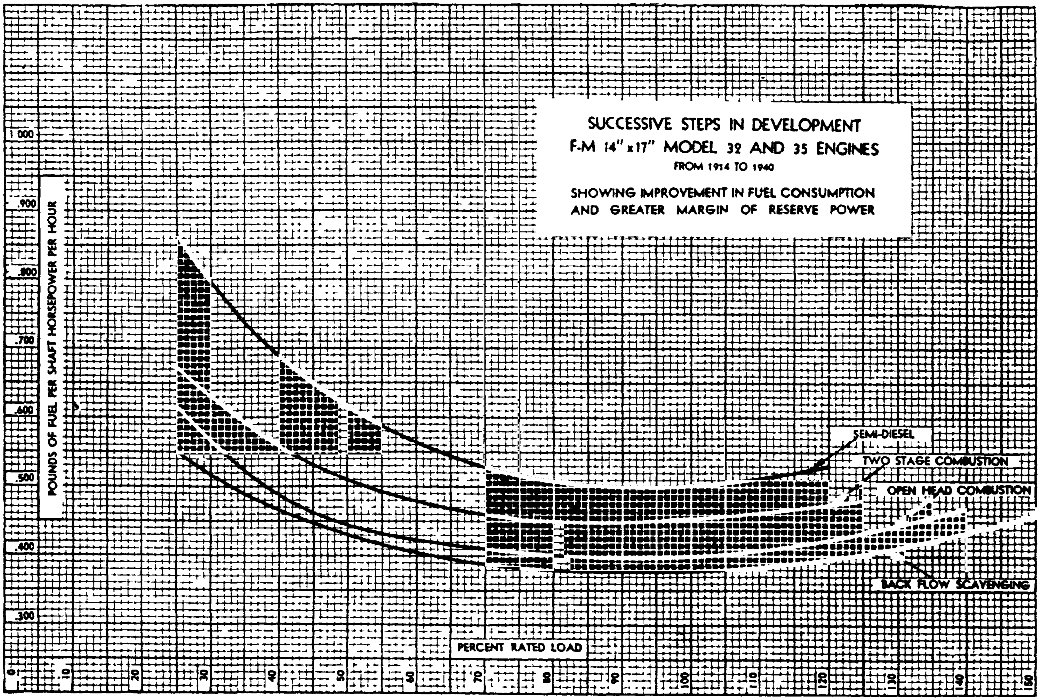


FIG. 2-1. Fairbanks Morse engines represent thirty years' development and perfection of basic-type two-cycle Diesel engines from the slow-speed, surface-ignition, semi-Diesel engine of the 1914 design to the modern, opposed-piston, uniflow-scavenging, and centrifugal-type blower.

improvements in Diesel design and construction in this country. The use of alloy metals is now an accepted practice, but such practices were first suggested by Europeans. Many combustion chamber forms and fuel injection systems originating in Europe have also been incorporated in American engines.

The use of supercharging to increase the output per cylinder size has been more widely developed in the United States than in Europe. Likewise, Americans are leading other countries

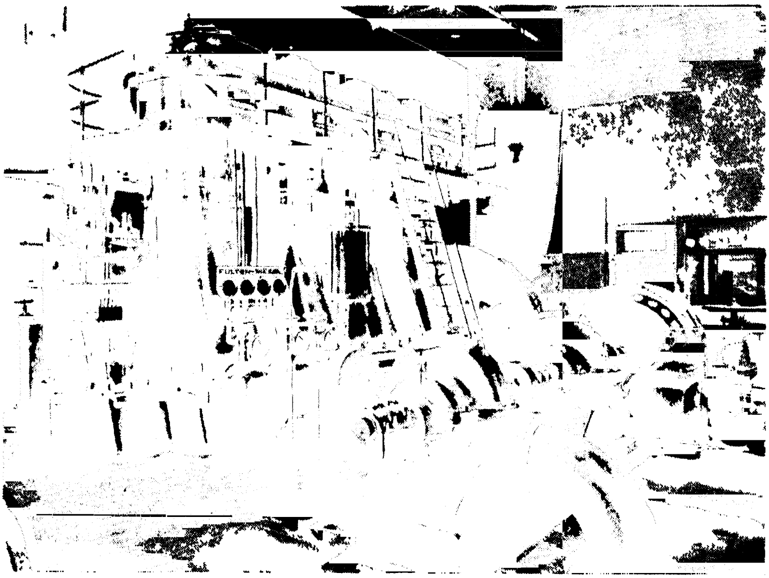


FIG. 2-2. Fulton air injection Diesel—slow-speed, A-frame construction, with water-cooled pistons. This engine represented the most conservative ideas of design of heavy-duty air injection engines of the early 1920's. It is rated at 750 hp and installed in the Los Angeles Water Works pumping station, in which there were at that time 4950 hp in Fulton Diesel engines for water pumping.

in the use of better accessories. The wide variety of conditions under which engines operate, and the problems of maintenance involved, make it necessary to give the best possible care to these engines in order to prolong the life and lower the upkeep cost if the Diesel engine is to complete with the gasoline engine in the high-speed fields.

Problems in the selection and application of the Diesel engine, when the right engine for the right job is being considered, involve many purely technical points of engine types

and designs, and, when mistakes of application are made, maintenance and operation problems are to be charged against failure to consider these points. A study of these technical points is required for a sound understanding of the problems involved in the selection of a Diesel engine and its proper application.

Factors in selection and application of Diesel engines. Certain facts should be considered with the view of eliminating as far as possible an improper selection and application of Diesel engines. These are:

1. The type. Although certain applications dictate certain types, the details of design, materials, and workmanship are usually more important. The record of the engine's performance and the reputation of the manufacturer are likewise to be considered.

2. Horsepower rating. Horsepower rating is the product of cylinder displacement, mean effective pressure, and revolutions per minute. Simply getting more power out of a given size of engine to lower the horsepower cost may shorten the life of the engine. Any engine should be rated as conservatively in speed and mean effective pressure as the prospective purchaser can afford. This simply means that it is desirable to have as many cubic inches of piston displacement per horsepower developed as the application will permit.

3. Design. The design should be as simple as possible. Too many complications are viewed with suspicion, and should not be accepted if avoidable. Every simplification is an advantage in reducing maintenance problems. Space, weight, and other factors are prescribed by the application, but the space and weight should not dictate the type or details of the design to be selected when other factors such as maintenance are considered.

4. Weight of engines. The weight of the engine is a function of the weight per unit of displacement, mean effective pressure, and revolutions per minute. The weight per unit of cylinder displacement, or heaviness of material construction, is the characteristic involved. This characteristic depends largely on the design of the engine. Air injection engines were heavier than solid injection; crosshead engines weighed more than trunk piston engines; and the 2-cycle engine was larger than the 4-cycle when it had the added weight of the scavenging pump. Weight that accomplishes no good purpose is not justified. Each

TABLE 2-1
GENERAL MOTORS ENGINE
GENERAL SPECIFICATIONS

	Model				
	1-71	2-71	3-71	4-71	6-71
Number of cylinders	1	2	3	4	6
Bore and stroke, inches	$4\frac{1}{4} \times 5$	$4\frac{1}{4} \times 5$	$4\frac{1}{4} \times 5$	$4\frac{1}{4} \times 5$	$4\frac{1}{4} \times 5$
Total displacement, cubic inches	70.9	141.8	212.7	283.6	425.4
Maximum speed, rpm	1600	2000	2000	2000	2000
Maximum bhp output at maximum speed	25	55	83	110	165
Maximum bhp output at 1200 rpm		40	60	80	120
Industrial rating at 1200 rpm	15	30	45	60	90
Maximum torque, lb-ft		178	262	350	525
at rpm	800–1200	800–1200	800–1200	800–1200	800–1200
Compression ratio, nominal	16:1	16:1	16:1	16:1	16:1
Piston speed ft/min @ 1200 rpm	1000	1000	1000	1000	1000
Number of exhaust valves per cylinder	2	2	2	2	2
Exhaust valve diameter, inches	1.564	1.564	1.564	1.564	1.564
Exhaust valve seat angle, degrees	45	45	45	45	45
Exhaust valve lift, inches	.375	.375	.375	.375	.375
Firing order: clockwise rotation	—	—	1-3-2	1-3-4-2	1-5-3-6-2-4
Firing order: counterclockwise rotation	—	—	1-2-3	1-2-4-3	1-4-2-6-3-5
Number of main bearings	2	3	4	5	7
Main bearing diameter, inches	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$
Main bearing length, inches	$2\frac{1}{4}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{1}{8}$
Crankpin diameter	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$	$2\frac{3}{4}$
Crankpin bearing length, inches	$1\frac{25}{32}$	$1\frac{25}{32}$	$1\frac{25}{32}$	$1\frac{25}{32}$	$1\frac{25}{32}$
Clutch size, inches	8	10	11	12	13
Shipping weight of basic engine (dry)* pounds	875	740	1450	1650	2000
Shipping weight of portable power unit, lb	1030	—	2500	2800	3200
Shipping weight of portable generator unit	1480	—	3350	3542	4300
Fuel consumption lbs/bhp hr	.50	.45	.45	.45	.45
Capacities					
Lubricating oil system, quarts	8	7	$12\frac{1}{2}$	$16\frac{1}{4}$	22
Fuel oil tank, gallons	None	—	33	43	63
Cooling water, quarts, engine only	3.6	5	$9\frac{1}{2}$	12	$21\frac{1}{2}$

* Engine weight includes: starting motor, governor, oil cooler, oil filter, fuel filter, generator, cooling fan, oil bath air cleaner, air intake elbow, and engine mountings.

type of engine has a normal weight, and, if this is exceeded, there is no compensating advantage.

Many applications require low unit weight. This low unit weight has been achieved by a number of methods. The cylinders are placed closer together; aluminum is used; and welded sheet steel replaces cast iron in the framing.

Changes in engine constructions. Other changes that departed from conventional engine construction have been made and are justified when the engine application requires it. For a number of years prior to World War II, design technique was changing so rapidly that it was difficult to separate experimental design from the practical.

The following are the features of practical design now widely accepted:

1. Vertical in-line cylinder arrangement and V types.
2. Mechanical injection.
3. Aluminum uncooled pistons in cylinder of 100 hp and larger, and cast-iron pistons for larger slow-speed engines.
4. Moderate brake mean effective pressures in unsupercharged engines.
5. Increased piston speeds.
6. Multiple inlet and exhaust valves.
7. Built-up welded alloy steel frames or tie-rod cast-iron frames.

The following features of design are less accepted practice:

1. The use of supercharging to increase the power output of the smaller cylinder and to avoid effects of valve size restriction to air and exhaust gas flow.
2. The use of individual fuel pumps or block pumps, open or closed nozzles, or common rail systems is still controversial.
3. The use of means to lower the weight of framing that interfere with accessibility, such as suspending the bearings and using valve seats fixed in the cylinder head.

World War II engines. At the present time the limit to the size of the 4-cycle, high-speed, light-weight engine with uncooled pistons is still unsettled, but there are engines with cylinders developing 125 to 150 hp. However, World War II witnessed the development of high-speed Diesel engines for submarines, surface craft, and industrial applications in outputs up to 125 hp per cylinder, weighing less than 20 lb per horsepower,

TABLE 2-2
CUMMINS DIESELS
GENERAL SPECIFICATIONS

	Model							
	A-400	A-600	H-400	H-600	HS-600	K-600	KO-600	L-600
Number of cylinders	4	6	4	6	6	6	6	6
Bore (inches)	4	4	4⅞	4⅞	4⅞	6¾	7	7
Stroke (inches)	5	5	6	6	6	9	9	10
Displacement (cubic inches)	251	377	448	672	672	1932	2078	2309
Compression ratio	18¼–1	18¼–1	17–1	17–1	14–1	14–1	14–1	14–1
Horsepower (maximum)	67	100	100	150	200	210	230	250
Number main bearings	5	7	5	7	7	7	7	7
Type main bearings	Steel-backed removable shells							
Area main bearing surface (square inches)	23.95	33.64	45.2	62.1	62.1	123.38	123.38	123.38
Connecting rods	Forged alloy steel—all models							
Connecting rod bearings	Steel-backed removable shells							
Crankshaft	Nickel molybdenum steel—all models							
Camshaft	Special chrome steel forging—all models							
Camshaft bearings—number	5	7	5	7	7	7	7	7
Cylinders	Cast en-block with removable, wet-type liners							
Valve location	Overhead—all models							
Valves	Intake and exhaust—heat-resisting alloy steel							
Pistons	Cam ground							
Piston pins	Full floating—all models							
Piston pin bearings	2 in piston—1 in rod							
Piston rings—compression	3	3	3	3	3	5	5	5
Piston rings—oil	2	2	2	2	2	2	2	2
Crankcase and cylinder block	Iron alloy, cast integral—all models							
Crankcase oil capacity (gallons)	2	3	4½	5	5	18	18	18
Lubrication	Force-feed to all bearings							
Engine cooling system capacity (gallons)	2	3⅛	4	5	5	15	15	15
Governor	Mechanical fly-ball type							
Fan drive	Double V-belt—all models							
Starting method	Electric	Electric	Electric	Electric	Electric	Air	Air	Air
Lube oil—above 60° F.	No. 30	No. 30	No. 30	No. 30	No. 30	No. 30	No. 30	No. 30
Lube oil—15° to 60° F.	No. 20	No. 20	No. 20	No. 20	No. 20	No. 20	No. 20	No. 20
Lube oil—below 15° F.	No. 10	No. 10	No. 10	No. 10	No. 10	No. 10	No. 10	No. 10
Oil change recommendations	Governed by type of service							

operating at rated speeds of from 700 to 1000 rpm, and occupying less than ½ cu ft of space per horsepower.

While a marked reduction in cylinder size and multiplication of cylinders to obtain greater output involve an increased number of working parts and complications that cause greater

maintenance cost and a loss in reliability, the following improvements are characteristic of World War II engines:

1. Supercharging.
2. Increasing the piston speed by lengthening the stroke and increasing the revolutions.
3. Shifting to 2-cycle design of improved types.
4. Using larger uncooled aluminum pistons and eliminating piston cooling.

Factors of weight and space. A number of major developments in the Diesel engine between World War I and World War II contributed importantly to weight and space savings:

1. The use of lighter reciprocating parts made possible the increase in piston speed and revolutions per minute.
2. Light, well-designed, and compact engine frames.
3. The replacing of air injection in marine engines by solid injection.
4. Many metallurgical developments in the use of ferrous materials as well as light alloys for stressed and unstressed engine members.
5. Increase of brake mean effective pressures by use of supercharging.

What are the significance and advantages of each of these features that contributed to savings in weight and space and how are they related to operation and maintenance?

Piston speed and revolutions. The design of the high-speed Diesel engine was made possible as a result of:

1. Decrease in connecting rod and piston weight.
2. Improvement in volumetric efficiency by increasing the size of valves.
3. Less weight of valve and valve gear.
4. Improvement in bearings of all kinds.

Decrease in piston weight. Less piston weight has been obtained in cylinders with bores up to 13 and 14 in. by the use of aluminum uncooled pistons. It is probable that larger uncooled pistons may be used. The practical cylinder output for the high-speed marine engine is now about 120 to 125 hp. Piston speeds are practically doubled over those of World War I, and the total weight of the engine is halved.

Valve area increase. The increase in piston speeds and revolutions per minute developed the need for increase in valve

area, on account of the excessive gas velocities produced by higher speeds, which cause wire-drawing through the smaller valves. The insufficient valve area would entail:

1. Loss of brake mean effective pressure due to back pressure at the exhaust valve.
2. Wire-drawing through the suction valve causes loss in volumetric efficiency.

In order to overcome such drawbacks of the small valves, the use of dual valves in high-speed, 4-cycle engines was adopted. However, the limit is two exhaust valves and two inlet valves. More than two of each have been used, but when this complication is added there are:

1. Cylinder head and valve drive complications.
2. The need for added power to actuate the valves.
3. Increase in troubles with valve overhaul and upkeep cost.

In some engines the inlet air velocity is around 10,000 ft per minute, which is about the maximum allowable; in fact, less than 8000 ft per minute is better practice. Exhaust valve velocities as high as 13,000 ft have been used, the velocities being measured at the valves. The loss in mean effective pressures as a result of wire-drawing of the air passing through the inlet air port can be serious. The inlet valves are often multiple and are larger than exhaust valves. The losses from reduced volumetric efficiency are far more serious than those that are produced by back pressure, and some engines have dual inlet and single exhaust valves. A number of 4-cycle engines use multiple inlet valves to increase breathing capacity for this reason. Another factor in the use of multiple valves is the fact that cooled exhaust valves used in older slow-speed engines are not so practical in high-speed engines, on account of the inertia of the weight of the water.

Less weight of valve and valve gear. The weight of valves and valve actuating gears had to be reduced when high rotative speeds were adopted, since the higher angular velocities of larger cams would have caused inertia forces, and these forces induce wear and chatter between the cams and rollers. Also, the excessive power to drive heavy valve gear and cam had to be eliminated to prevent abnormal wear of cam and rollers. The wear would be rapid with heavy gear at high rotative speeds.

Bearing pressures. It was previously pointed out that mechanical or solid injection as compared with air injection produced more severe maximum pressures with its shocks, thus

increasing the maximum load on bearings. Although bearings have been much improved, this matter of high maximum pressures has led to increased bearing pressures.

A rough ratio between piston top area and crankpin bearing area is 2.5 to 1 to 2 to 1. Main bearing areas range between 2.5 to 1 and 3 to 1, while wrist pin area is usually 5 to 1 to 4 to 1.

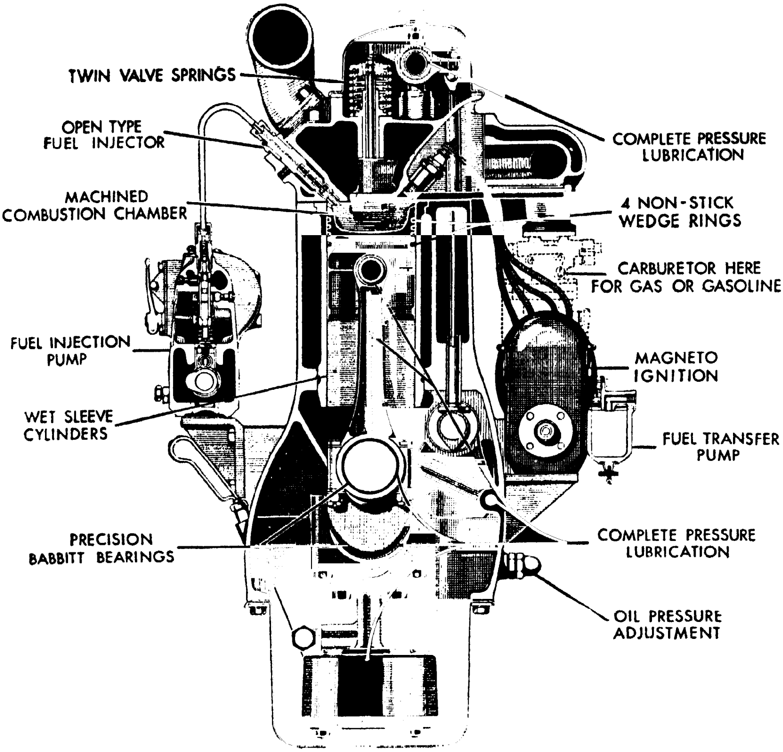


FIG. 2-3. Cross-section through the Waukesha multi-fuel (Hesselman) engine. This design represents numerous improvements and advancements in automotive and marine engines; the engine is classed as a spark-ignition, fuel-injection engine, and burns kerosene, distillate, and Diesel fuels. It is used for tractors, trucks, and similar applications.

This requirement depends upon the nature of peak pressures, but it is always desirable to reduce maximum bearing pressures. A check of engine specifications will show how this question has been considered by the designer. High bearing pressures cause a great deal of maintenance work on the Diesel engine.

Frame construction. Engine builders have been able to reduce the ratio of the total prismatic volume to total piston

displacement. This reduces space and weight requirements and involves making the engine framework lighter and stronger. The essential changes that were made in the construction of the

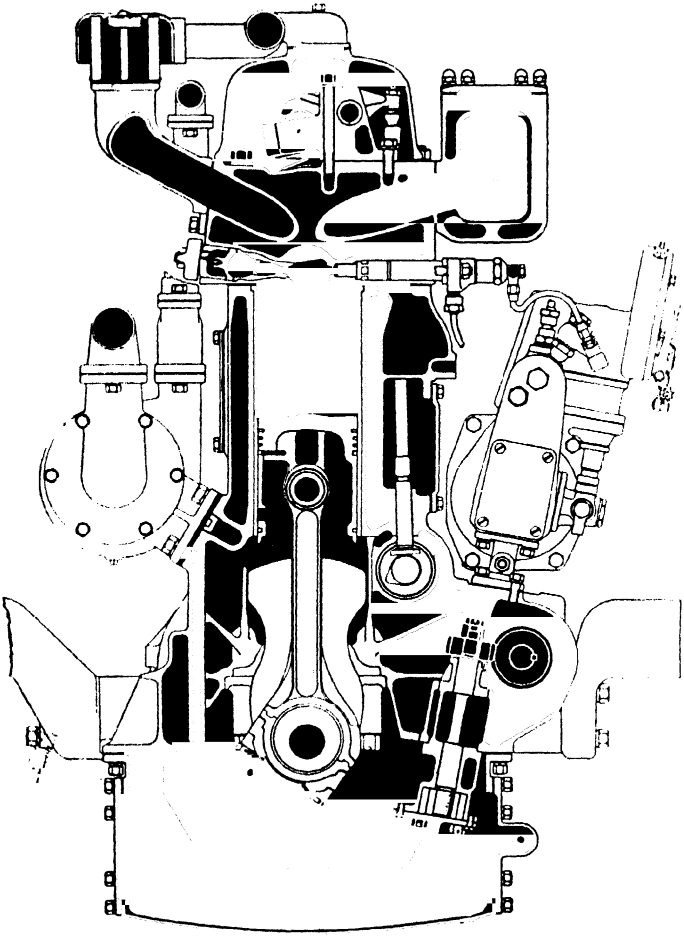


FIG. 2-4. Mack-Lanova (air-cell) marine Diesel engine. This engine is typical of high-speed marine and automotive designs. Several American engines employ the Lanova combustion system. It has "dry"-type cylinder liners.

engine concerned cylinder arrangement. Some of these changes are:

1. V-type frame. Normally engines are built with vertical cylinders in line. Some noteworthy World War II engines

use the V-type frame; and one, the radial aircraft style of frame. The V type enables the builder to attain the reduced over-all height so desired in the locomotive and the submarine types of engines. The V-type engine also permits multiplying the cylinders without unduly increasing the over-all length, ultimately increasing the power from a given unit. Engines with small cylinders and aluminum pistons can be built for greater output by using the V style to increase cylinders, thus avoiding the technical difficulties involved with larger cylinder sizes and limits to the size of aluminum pistons.

Many main-line Diesel-electric locomotive engines use the V-type arrangement to obtain reduction in headroom and the reduction in the generator size and weight made possible by high rotative speed. The General Motors marine engine, whose characteristics are listed on page 50, represents a successful V-type light-weight engine in large cylinder sizes, used for locomotive and marine propulsion. There are few other examples of the V-type arrangement in the field, and it is believed that the multiple, vertical in-line arrangement will continue to dominate.

A study of engine frame construction shows that some frame designs are inherently light and others inherently heavy. Our World War I engine of the light-frame type was a heavy cast-steel crankcase and a housing with cast-steel cylinders cast en-block or bolted to give stiffness. This frame was unnecessarily heavy because of the poor transmission of tensile stresses.

2. *Tie-rod construction.* High-speed Diesels of the 4-cycle type built in smaller sizes have the box type with en-block cylinders or bolted up heads and tie-rods. The tie-rod has been widely used for both large and small engines, because, when used to reduce frame weight, it permits:

(a) The suspension of the main bearings. The heavy bedplate used for main bearings on the World War I engine is eliminated.

(b) The heavy cast-iron water jacket is also eliminated and in its place is a light sheet-metal water jacket housing for present day high-speed engines.

(c) The wall section of the frame is quite thin now since the tie-bolts are able to take all tensile stresses.

The tie-rod construction, now widely used for larger types of light-weight engines and marine and locomotive applications,

makes possible the suspension of the main bearings. The elimination of the heavy bedplate was a very material reduction in the weight of the engine.

3. Cylinder liner and light water jacket. Since World War I, the replacement of the heavy housing by the use of the light metal water jacket has been an important development. The integral iron cylinder or heavy cast-iron liner of the engines of World War I days has given way to a thin section alloy liner. This light wall section liner may be a forged alloy steel instead of a casting. It is a significant new development found in the present engine types illustrated herewith.

4. Valve cages. A material savings in engine weight resulted from the elimination of the valve cages. Formerly large engines with the fitted cages in the cylinder heads for valves were favored because the arrangement permitted the rapid removal of the valve without removing the cylinder head. A particular disadvantage was that the large cages reduced the valve area possible and complicated the head, but, since the modern engine of the high-speed type requires maximum possible valve area, valve cages cannot be used. The direct seating arrangement of the valves permits more effective valve cooling than does the cage arrangement, and the cylinder head is less complicated.

The use of suspended bearings is more practical with the more compact frame now used, since the tie-rods themselves serve as bearing bolts. This also results in reducing the width of the engine and the cylinder center-line distances. Present engines are scarcely half as long as World War I engines.

5. Objections to light frames. The changes in frame designs that resulted in a saving in frame weight and space had disadvantages, and certain advantages of the older style construction were sacrificed.

1. When the tie-rods were used and main bearings suspended, the accessibility of the bearings was reduced. The removal of the crankshaft became more difficult. This arrangement also made adjustment of the compression complicated, since the simple arrangement for removal of shims from the foot of the rod was interfered with.

2. One-piece tie-rods make it difficult to dismantle the engine in restricted places, such as locomotive and marine applications. The one-piece tie-rod has been eliminated in most American designs.

3. The routine dismantling and overhauling of the valves is

TABLE 2-3

MACK-LANOVA DIESELS

Type	Truck Engines			Bus Engines			Marine Engines				
Model	END405	END519	END605	END457	END519	END605	END457Y	END457W	END519Y	END605Y	END605W
Combustion system	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova	Lanova
Cycle	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke	4-stroke
Number of cylinders	Six	Six	Six	Six	Six	Six	Six	Six	Six	Six	Six
Bore and stroke	$4 \times 5\frac{3}{8}$	$4\frac{3}{8} \times 5\frac{3}{4}$	$4\frac{5}{8} \times 6$	$4\frac{1}{4} \times 5\frac{3}{8}$	$4\frac{3}{8} \times 5\frac{3}{4}$	$4\frac{5}{8} \times 6$	$4\frac{1}{4} \times 5\frac{3}{8}$	$4\frac{1}{4} \times 5\frac{3}{8}$	$4\frac{3}{8} \times 5\frac{3}{4}$	$4\frac{5}{8} \times 6$	$4\frac{5}{8} \times 6$
Piston displacement (cu in.)	405	519	605	457	519	605	457	457	519	605	605
Hp @ gov. rpm max.	107 @ 2200	131 @ 2000	144 @ 2000	122 @ 2200	131 @ 2000	144 @ 2000	115 @ 1950	110 @ 1800	120 @ 1800	140 @ 1800	125 @ 1800
Continuous sustained duty	—	—	—	—	—	—	—	70 @ 1500	—	—	100 @ 1500
Max. torque lb/ft @ rpm.	308 @ 1200	382 @ 1300	455 @ 1100	355 @ 1200	382 @ 1300	455 @ 1100	355 @ 1200	280 @ 1300	382 @ 1300	455 @ 1100	410 @ 1000
Cylinders and crankcase cast	En-block	En-block	En-block	En-block	En-block	En-block	En-block	En-block	En-block	En-block	En-block
Type of cylinder liners	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Dry
Cylinder heads cast in	Three	Three	Three	Three	Three	Three	Three	Three	Three	Three	Three
Piston material	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum	Aluminum
Crankshaft material	Drop-forged, low-carbon steel			Drop-forged, low-carbon steel			Drop-forged, low-carbon steel				
Treatment	Case-hardened			Case-hardened			Case-hardened				
Number of counterweights	12	12	12	12	12	12	12	12	12	12	12
Vibration damper	Lanchester	Lanchester	Lanchester	Lanchester	Lanchester	Lanchester	—	—	—	—	—
Main bearings, type	Thin-shell, copper lead			Thin-shell, copper lead			Thin-Shell, copper lead		Thin-shell,		
Number	7	7	7	7	7	7	7	7	7	7	7
Diameter and length	$3\frac{1}{2} \times 10\frac{31}{32}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 10\frac{31}{32}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 10\frac{31}{32}$	$3\frac{1}{2} \times 10\frac{21}{32}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 11\frac{1}{2}$	$3\frac{1}{2} \times 11\frac{1}{2}$
Camshaft bearings	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt	7-babbitt
Timing drive	Drop-forged, case-hardened gears			Drop-forged, case-hardened gears			Drop-forged, case-hardened gears				
Exhaust valve seats	Stellited Niferrite			Stellited Niferrite			Stellited Niferrite				
Valve ports	Individual	Individual	Individual	Individual	Individual	Individual	Individual	Individual	Individual	Individual	Individual
Injection pump, make	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch
Mounting	Flange	Flange	Flange	Flange	Flange	Flange	Flange	Flange	Flange	Flange	Flange
Drive	Gear	Gear	Gear	Gear	Gear	Gear	Gear	Gear	Gear	Gear	Gear
Timing	Synchrovance variable automatic			Synchrovance variable automatic			Fixed	Fixed	Fixed	Fixed	Fixed
Injection nozzles, make	Excello	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch	Bosch
Governor, make	Bosch	Pierce	Bosch	Bosch	Pierce	Bosch	Bosch	Bosch	Pierce	Bosch	Bosch
Starting heaters	Electric resistance type in intake ports			Electric resistance type in intake ports			Electric resistance type in intake ports (extra charge)				
Temperature control	Cold-circulation type			Cold-circulation type			Cold-circulation type				
Cooling for marine engines	—	—	—	—	—	—	Dual type, fresh water in closed system and sea water				
Heat-interchanger	—	—	—	—	—	—	Harrison unit with large capacity expansion tank				
Fresh water pump	—	—	—	—	—	—	Centrifugal bronze rotor on nitrided steel shaft				
Salt water pump	—	—	—	—	—	—	Gear type, double shaft with separate driving gears				
Lube oil filter	Cotton bag type			Cotton bag type			Cotton bag type				
Lube oil cooler	—	—	—	—	—	—	Built onto sea water pump				
Secondary fuel filters	Two, cotton waste and filter paper			Two, cotton waste and filter paper			Two, cotton waste and filter paper				

rendered less convenient with the use of heads with integral valve seats instead of separate cages. More time is involved in maintenance work as it requires the lifting of the cylinder head instead of easy removal of the cages.

Metallurgical advances. All present-day engines contain metallurgical improvements and use improved ferrous materials in the construction of the engine. Special alloys are also used.

1. Improvement in the life and reliability of valves has been made by use of high-temperature alloy steels.
2. Better qualities of ferrous alloy for rings and liners result in decreased wear on these parts.
3. Alloy steel shafts, tie-rods, and connecting rods have greater strength.
4. Alloy semisteel for cylinder heads, pistons, and cylinder liners as well as for valves is now used and affords greater strength and reliability.
5. Welded steel plate, a recent introduction, is used for fabrication of the frames of large engines and contributes to the lower cost of manufacture.
6. An outstanding improvement is the use of pearlitic iron or semisteel for liners, frames, cylinder heads, and other light parts, although the practice is not universal.

TABLE 2-4
GUIBERSON DIESELS
General Specifications

	Model T-1020	Model T-1400
Cylinder arrangement	Single-row radial	Single-row radial
Cooling media	Air	Air
Total piston displacement	1021 cu in.	1402 cu in.
Bore	$5\frac{1}{8}$ in.	$5\frac{3}{4}$ in.
Stroke	$5\frac{1}{2}$ in.	6 in.
Hp	250	350
Rpm	2200	2200
Compression ratio	14.5:1 max.	16:1 max.
Propeller	Direct drive	Direct drive
Rotation, facing power take-off end	Counterclockwise	Counterclockwise
Injection system	Plunger-type injector, solid	Plunger-type injector, solid
Injection pressure	—	2500 psi
Thermodynamic cycle	Diesel, 4-stroke	Diesel, 4-stroke
Dry weight (engine less accessories and starter)	714 lb	1100 lb max.
Over-all dimensions (diameter)	$45\frac{1}{2}$ in. max.	52.75 in. max.
Length with starter	37 in. max.	42.75 in. max.

Alloy steel rods. The adoption of alloy steel tie-rods to relieve the cast frame of tension permits the use of thin-wall cast-iron frames or light frames of welded steel construction. When the frame itself is relieved of tensile stresses, much thinner and lighter sections can then be used. The result is that the frames may be and are made of very thin sections of pearlitic, or partial pearlitic cast iron. Formerly, the heavy box frame made of cast iron was employed to absorb the tensile stress. Since it was impossible in these complicated structures to determine what part of the total stresses each wall absorbed, the frames were always made unusually heavy as a factor of safety. The cast iron was weak under tension and very resistant to compression, whereas cast steel is stronger in tension but difficult to cast in thin sections. The tie-rod proved to be the answer to some design requirements.

Disadvantages of supercharging. Diesel engines of the 4-cycle type may be supercharged up to 40 to 50 per cent overload without seriously increasing the exhaust temperatures. Supercharging was a logical adoption to decrease space and weight and increase output, but the value of supercharging for various applications is still being debated in some quarters. Its value for intermittent overload is unchallenged. Certain drawbacks may be enumerated here:

1. The added blower and its drive is a complication. Supercharged engines are better than the double-acting 2-cycle type.

2. Many builders contend that it is better to add cylinders to increase the power rather than blowers that consume power if the weight-power ratio is to be kept down.

3. It may be contended that while mean temperatures may be about the same in supercharged as in unsupercharged engines, the end temperatures are somewhat greater, which means that heat stresses are more severe, a factor in shortening engine life, and increasing the maintenance work.

4. The supercharger is not a step in the direction of simplicity. Piston speeds are limited by insufficient valve area and supercharging helps to offset this, but the velocity of the exhaust gas through the exhaust valve is increased. This results in considerable back pressure and valve stem troubles, so that means must be found to avoid this. The exhaust turbine or turboblower causes higher back pressure and increases exhaust valve difficulties from a maintenance standpoint.

5. The design of a simple and reliable scavenging blower

which is economical in weight and space is another problem that has long engaged the attention of the builders. The positive displacement-type air pump of the Roots design rather than the heavy piston type has many advantages. Electrically driven centrifugal blowers are used on larger installations but not on smaller light-engine applications.

TABLE 2-5
MAIN PROPULSION ENGINE—MARINE
General Motors

Characteristics	
Type	V engine
Cycle	4-stroke
Bore and stroke	$9\frac{1}{2} \times 12$ in
Number of cylinders	16
Rated engine speed	900 rpm
Full load at rated speed	1800 bhp
Bmep	116.4 psi
Piston speed	1800 ft per min
Piston displacement	13,609 cu in
Compression ratio	15.05:1
Compression pressure	655 to 710 psi
Firing pressures	980 to 1015 psi
Fuel consumption	0.378 lb per bhp
Starting air pressure	500 psi
Air-fuel ratio	42 to 1
Blower type	Turboblower
Type of fuel injector	Unite injector
Injector nozzle pressure	2200 psi
Injector timing	5° before TDC
Cylinder relief valve set	1350 psi
Maximum exhaust temperature	750° F
Lubricating oil temperature	160° F
Cooling water temperature from engine	100° F to 160° F
Overspeed trip set at	990 rpm
Scavenging air pressure	0 to 5 lb
Bhp per cu in. displacement	
Weight of engine lb per bhp	

Advantages of supercharging. Some engineers claim that supercharging does not increase cylinder temperatures, and that 50 per cent power increase is obtained by an increase of 10 or 15 per cent in weight and space for the supercharger. Supercharging engines where full power is used intermittently is an established practice. Locomotives and marine installations are examples of this application. Supercharging is also claimed as a cure for insufficient valve area and its result, the reduction in volumetric efficiency at higher rotative speeds.

A recent report on *Supercharging Diesel Engines* was delivered at the Metropolitan Section of SAE by Mr. Richard Herold of Sulzer Bros., Ltd. His paper was entitled "Supercharged Two-Stroke Cycle Diesels," in which he says that supercharging (2-cycle engines) is not limited to fast-running opposed piston engines, that it has been applied just as well to medium-speed

TABLE 2-6
MAIN PROPULSION ENGINES—REVERSIBLE
Fairbanks Morse

Characteristics	
Type	Opposed piston
Cycle	2-cycle
Bore and stroke	$8\frac{1}{2} \times 10$ in.
Piston displacement	10,370 cu in.
Number of cylinders	10
Rated engine speed	800 rpm
Full load, at rated speed	1800 hp
Bmep at rated load and speed	89.7 psi
Compression ratio	14:1
Compression pressure	550 to 675 psi
Firing pressures	700 to 1200 psi
Fuel consumption, 720 rpm, 1600 hp	.374 lb per bhp per hr
Starting air pressure	250 psi
Blower type	Root's positive displacement
Scavenging air pressure	2 to 5 lb
Type of fuel injector	Bosch—solid injection
Injection nozzle discharge pressure	3000 psi
Maximum exhaust temperature	750° F
Cylinder relief valve set	2,000 psi
Maximum cooling water temperature	160° F from engine
Maximum lubricating oil temperature	180° F from engine
Weight of engine	33,180 lb
Piston speed	1200 ft per min
Bhp per cu in. displacement	
Weight lb per bhp	

single-piston engines, having bores from $3\frac{1}{2}$ to 21 in., by supercharging pressures up to 85 psi abs. Mean effective pressures two or three times those of atmospheric engines have been obtained. As the supercharging pressure increases, the importance of the blower and turbine increases, too. In the conventional engine, the scavenging compressor absorbs 6 to 9 per cent of the engine power. When the charging pressure is raised to 30 or 50 psi, the power absorbed by the compressor is around 25 to 40 per cent of the engine output. A two-stage axial compressor with intercooler may have a combined efficiency

well above that of the two stages. This arrangement increases the weight and cost. Thus the limits of supercharging may well be determined by economics rather than by thermodynamics. It is interesting to note that Herold reports that with supercharger efficiencies now available, Sulzer engines of high output

TABLE 2-7
MAIN PROPULSION ENGINE—REVERSIBLE
Hamilton, H.O.R. Marine Engine

Characteristics	
Type	Double-acting, reversible
Cycle	2-stroke
Bore and stroke	$9^{1}/_{16} \times 13^{3}/_{8}$ in.
Number of cylinders	9 upper and 9 lower
Engine speed	700 rpm
Full load, at rated speed	1800 hp
Bmep at full load and rated speed	68.7 psi (avg. both cylinders
Compression ratio (upper and lower)	14.3:1 and 12.9:1
Compression pressure, upper cylinders	500 to 580 psi
Compression pressure, lower cylinders	430 to 510 psi
Firing pressures, upper cylinders	880 to 920 psi
Firing pressures, lower cylinders	730 to 770 psi
Fuel consumption	0.42 lb per Bhp per hr
Starting air pressure	600 psi
Air-fuel ratio	40 to 1
Blower, separate type	Not connected to engine
Blower capacity	8000 cu ft per min
Type of fuel injector	Bosch—solid injection
Injector nozzle discharge pressure	3800 psi top, 3300 psi bottom
Injector timing, upper cylinders	17° before TDC
Injector timing, lower cylinders	24° before BDC
Cylinder relief valve setting	1050 to 1100 psi
Scavenging air pressure	4 to 5 psi
Maximum exhaust temperature	750° F
Lubricating oil temperature	180° F
Overspeed trip set	750 rpm
Weight of engine	26,500 lb
Piston displacement, cu in.	—
Bhp per cu in. displacement	
Weight lb per bhp	—

density are built with supercharger pressures of 28 to 40 psi abs, yielding a brake mean effective pressure of 140 to 215 psi. An engine supercharged with two atmospheres or 28.4 psi has an exhaust outlet pressure of 8.4 psi. The inlet pressure of the exhaust gas turbine is practically identical with the supercharging pressure.

It was also found that raising the supercharging air pressure to 30 psi gave the remarkable result of increasing the brake mean effective pressure more than 100 per cent. This suggested still higher supercharging pressures. This was tried out and it was found that the gain in brake mean effective pressure decreases as the supercharging is increased further, owing to the limitations imposed upon the engine by thermal stresses.

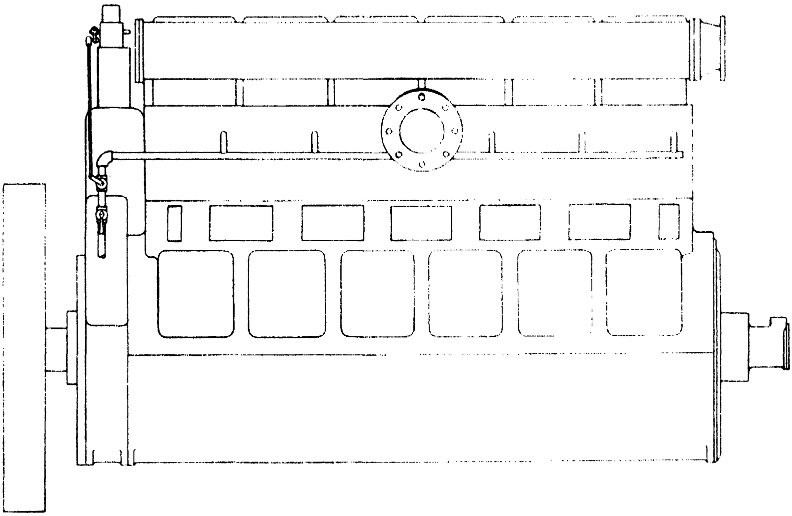


FIG. 2-5. This diagrammatic line drawing of a Diesel engine from the exhaust side shows how the new Cooper-Bessemer dual-fuel development is assembled to the engine. With this new improvement, any engine built as a Diesel can be fitted to operate on gas except spark-ignited gas engines. To convert the oil engine to a gas engine, it is merely necessary to remove the fuel oil system from the governor, set it to the minimum position, and connect the governor to the gas-regulating valve. All this is accomplished automatically by the turning of a valve.

An interesting development was observed by Mr. Herold in this connection. At supercharging pressures of 70 to 85 psi, depending on the blower and turbine efficiency, the ever-increasing power required by the blower becomes equal to the power developed by the engine. The power delivered by the turbine also becomes as high as that of the engine. Then it is immaterial whether it is the Diesel or the turbine driving the blower. If the blower is driven by the Diesel, the turbine may well be attached to the group. The turbine also becomes the producer of useful power.

Gas-Diesel development. A new development in the Diesel field that enables the operator to use either gas or Diesel fuel has been developed by the Cooper-Bessemer Corporation. The engine gives 25 per cent better fuel consumption than a regular spark ignition gas engine. The engine operates on a wide variety of fuels including fuel oil, natural gas, coke oven gas, sewage gas, and refinery by-products.

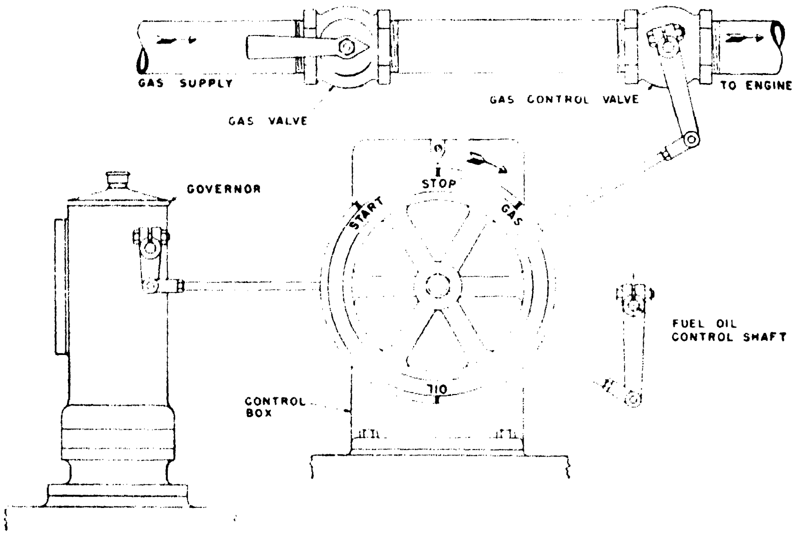


FIG. 2-6. Here is an enlarged view showing the method by which the controls are utilized to change from use of oil to gas, or vice versa, or even to maintain a mixture of the two where there might be insufficient gas to pull a load. It is possible to arrange it so that the engine automatically goes from one to the other as the gas availability fluctuates. Hundreds of installations at places where the future of the fuel supply is uncertain could solve their operating problem with this new method of control.

The conversion from a liquid to a gas fuel is accomplished without exchanging major parts of the engine. The engine is a Diesel engine and not a gas engine. It is fitted to operate on gas when this is desirable. The operation is as follows:

When operating as a Diesel engine, if necessary to admit gas in the intake air, the governor immediately reduces the amount of fuel oil to compensate for the additional heat units introduced with the gas. The fuel oil injection is reduced to the desired minimum and then the percentage of gas is governed according to the load. The system is shown in Fig. 2-5. In actual

practice, it is evident that the Diesel engine is converted to gas burning by removing the fuel oil system from the governor.

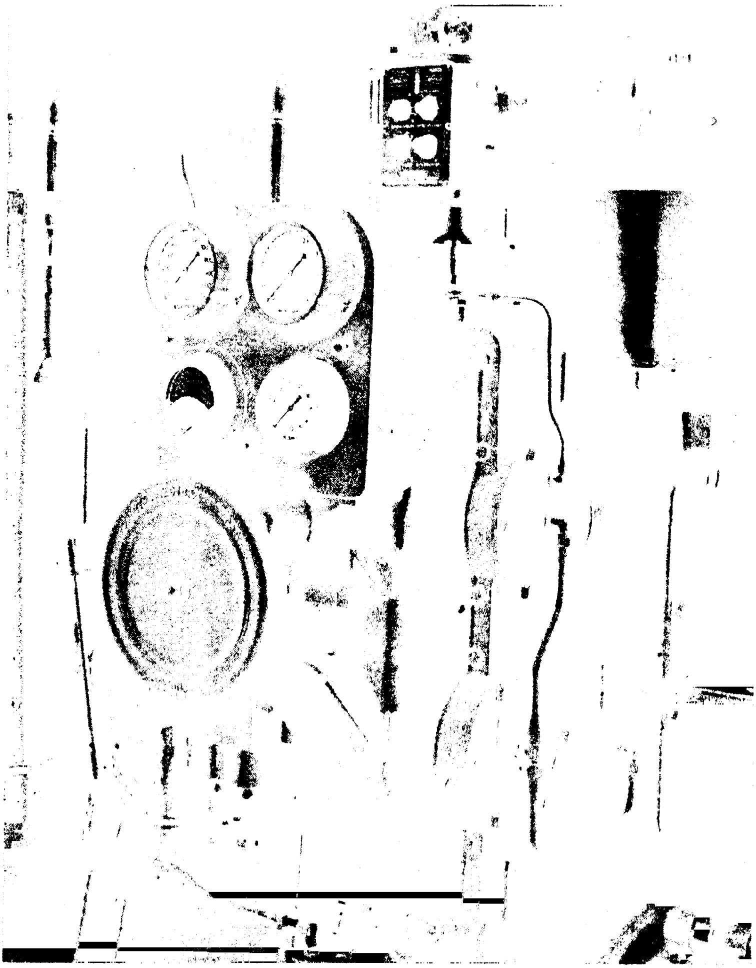


FIG. 2-7. This view of the control end of a Cooper-Bessemer Type JS.-8 Diesel engine showing the gas-oil operating mechanism. This engine is rated at 675 hp at 400 rpm.

and setting it in the minimum position to inject a pilot quantity of Diesel oil, at the same time connecting the governor to the gas regulating valve.

In applications where the gas availability varies frequently, such as a sewage disposal plant, it is desirable to be able to

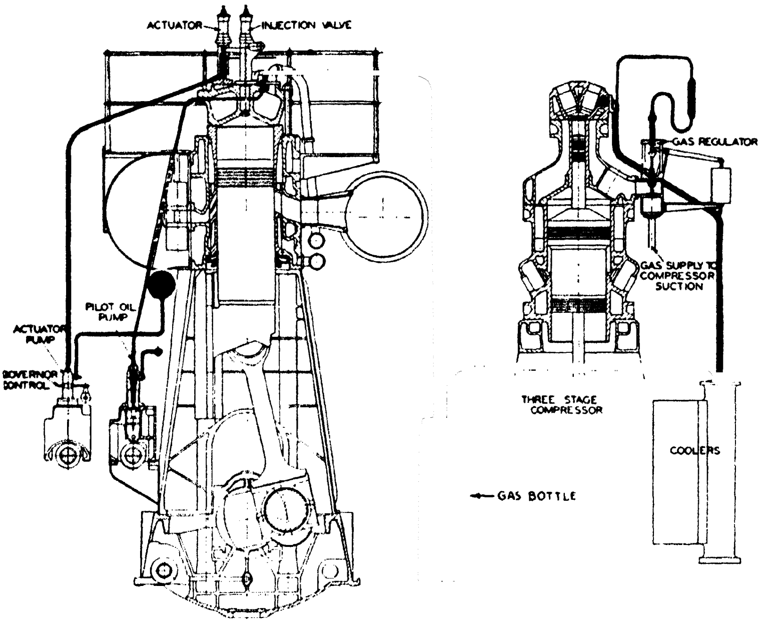


FIG. 2-8. Nordberg convertible gas-burning Diesel, showing details and piping. Several of these engines have been installed at the Lubbock, Texas, municipal plant and other places in the Southwest, using low-cost natural gas as fuel, at a fuel cost much below that for Diesel oil.

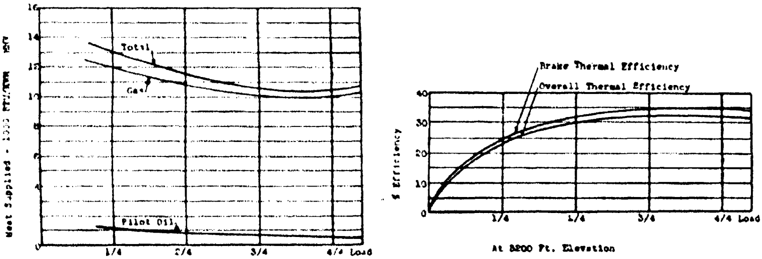


FIG. 2-9. Curves showing results of field tests at Lubbock, Texas, on an 8-cylinder 21″ engine, giving the heat supplied by pilot oil and gas at various loads, and total Btu; also brake and over-all thermal efficiencies.

change instantly from oil to gas or vice versa, or even to maintain a mixture of the two when there is insufficient gas to pull the load.

In order to do this, a form of control, as shown in Fig. 2-6, was developed that enables the operator to change from one fuel to the other, or even to arrange for the engine to go automatically from one to the other as the gas availability fluctuates. Fig. 2-7 shows a 675-hp Cooper-Bessemer engine with this gas-oil operating arrangement installed on the control end.

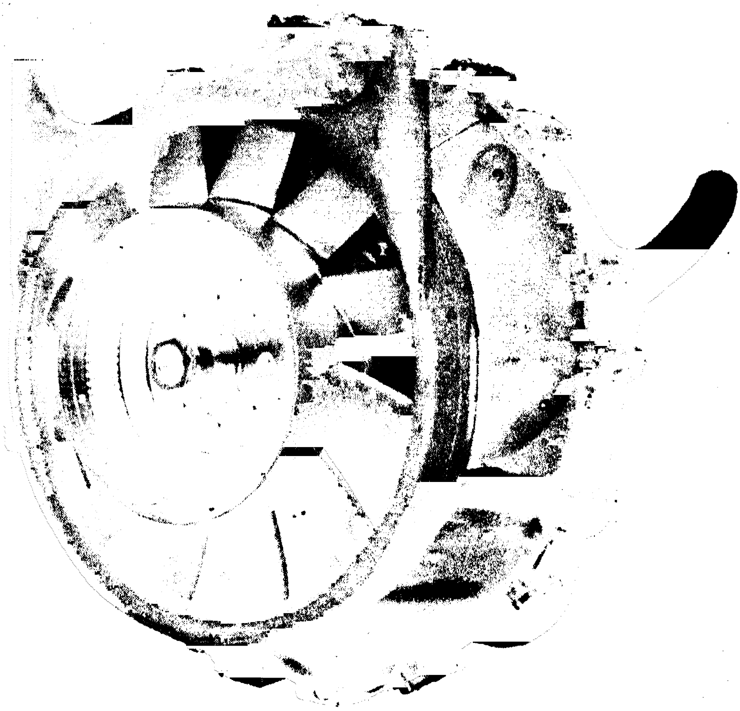


Fig. 2-10. Guiberson T-1020 Series 4 Diesel engine—$\frac{3}{4}$ left front view. Radial, 9 cylinder, air-cooled, aircraft type, adapted to use in U. S. Army tanks. The engine details are shown in the table of specifications.

The usual means of conversion of other engines is change of pistons, or cylinder heads, or by means of spacers under the cylinder heads which will increase compression space when converting to gas. Such a conversion also includes changing from an electrical ignition system to a fuel oil injection system, or vice versa. All of these methods have been used in the past, and hundreds of Diesel engines or gas engines have been installed

and used as convertible units on account of some uncertainty of the future supply of a particular fuel.

This Cooper-Bessemer method of burning gas on the Diesel cycle conserves gas and provides a simplified method of conversion, and at the same time provides for reconversion to the original fuel at any time. This application is of great impor-

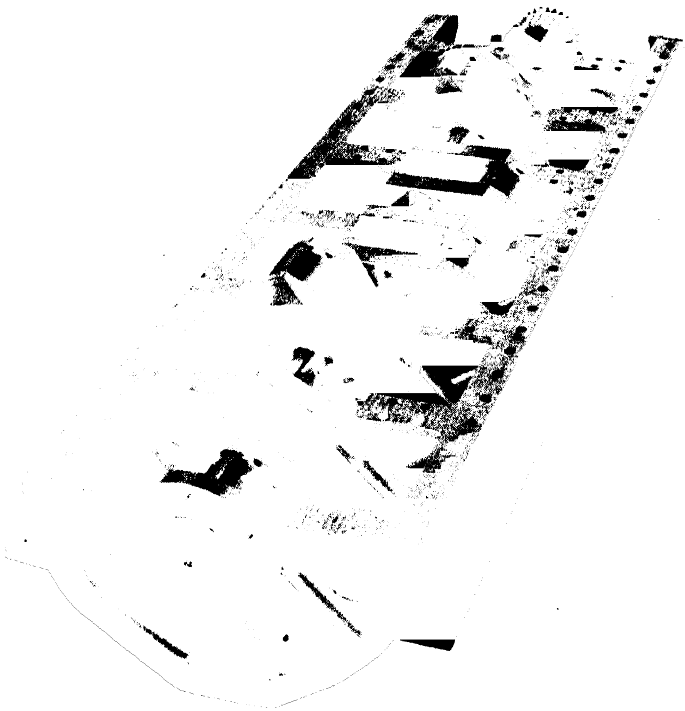


FIG. 2-11. DeLaVergne early 1930's design, high- and medium-speed Diesel, showing features of engine base and crankshaft construction.

tance for oil-field compressor engines, which frequently must be installed where the gas supply is not predictable.

Principle of operation. The gas is admitted with the intake air and is comparatively free of any evidence of preignition, as might be expected. The temperature of ignition of a perfect mixture of air and natural gas is actually above the temperature reached with 400 lb compression. The range of an explosive

mixture even for spark ignition is very narrow, and this engine operates with a lean mixture that is not too easily ignited; the air-gas ratio at full load is still a very lean mixture. At full loads the engine would not fire without a pilot quantity of oil injected with gas. The use of a pilot quantity of fuel oil when operating on gas is a distinct advantage. Using both fuels at

FIG. 2-12 Cylinder-head construction for medium-speed Diesel engine, early 1930's design by DeLaVergne

the same time has been mentioned. Actually better fuel economy should be possible when using two fuels simultaneously. The disadvantages of excess air as pointed out in Chapter 1 can be overcome in part when using some gas mixed with the air. The gas should contribute to better mixing and help to perform some of the mechanical functions performed by pure air. Less ignition lag and more uniform combustion is doubtless realized from this development. There is also the possibility of using a

Fig 2-13 Cross-section of Hendy Series 20 6-cylinder marine Diesel This is a 4-cycle, single-acting engine of cast-iron monoblock construction, with underhung crankshaft, removable cylinder liners, overhead cam shaft, unit injection, and auxiliary drive arranged at either end Attached auxiliaries are air compression, scavenging and pressure lube oil pump, centrifugal water pump, fuel transfer pump, fuel filter and instrument board

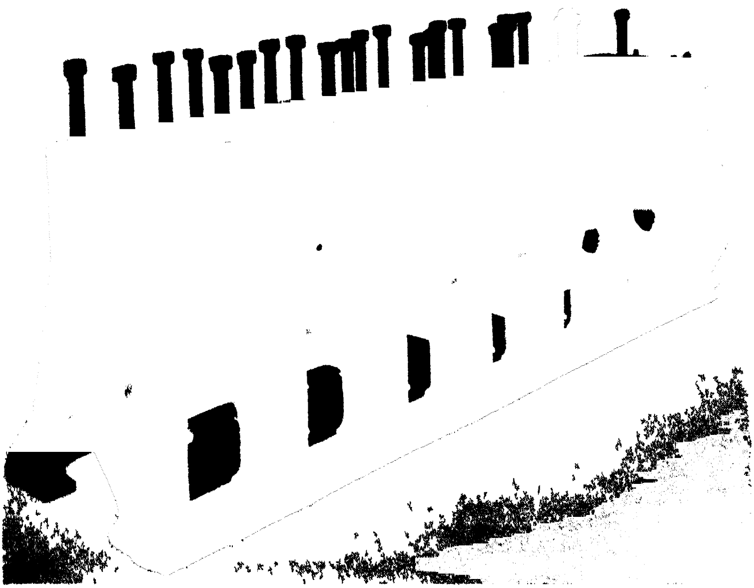


Fig 2-14 Early design of DeLaVergne medium-speed Diesel (800 rpm), showing box-frame construction and removable liners

heavier, cheaper grade of fuel oil to supplement high-priced natural gas for some city installations.

Gas injection Diesel. Another development of recent years is the gas injection Diesel engine using an air injection type of Diesel engine. Such an engine compresses air only, and the gas is injected on top dead center by the injection air compressor at a pressure of approximately 1000 psi. This engine uses pilot oil injection even though it would operate without it. This engine was developed by Nordberg Manufacturing Company, and many large installations have been made in the Southwest.

QUESTIONS

1. What items are considered in selecting an engine for any particular application?

2. The horsepower output of an engine is a function of what three factors?

3. What Diesel engine applications require low weight per horsepower?

4. Name the features of present-day practical designs.

5. What general features of modern designs are still unsettled, or less well established?

6. What special features were embodied in World War II engines to obtain greater output per engine weight and space requirements?

7. What other design features contributed to weight and space saving in modern designs?

8. What special steel and steel forgings are used for the crankshaft and cam shaft of such high-speed engines as the Cummins?

9. What kind of valve metal is used?

10. What piston speeds were employed for the high-speed engines, the specifications of which are given in the tables?

11. What had to be done to increase or improve the volumetric efficiency of the high-speed engine?

12. Why was valve area increased? What results from the use of insufficient valve area when piston speed and revolutions are increased?

13. What is the optimum air velocity through the inlet valve?

14. What maximum exhaust valve velocities have been measured?

15. How do some 4-cycle engines increase breathing capacity?

16. Are water-cooled exhaust valves practical in high-speed engines?

17. What is a rough ratio between piston top area and crankpin bearing area? Wrist pin or piston pin area?

18. What is the advantage of the V-type cylinder arrangement?

19. What are tie-rods? What are the disadvantages or objections to their use in the ultra-high-speed engines?

20. What metallurgical improvements are employed in the construction of the Mack-Lanova engines?

21. What is the name of the metal used in the exhaust valve seats?

22. What were the disadvantages of the valve cages and why did their use in the high-speed engine prove impractical?

23. What were their advantages in the older engines?

24. What metallurgical advantages are incorporated in the present-day engines?

25. What is the piston displacement of the Guiberson T-1020 radial aircraft Diesel engine? What is the number of cubic inches per horsepower rating? What is the weight per unit horsepower? What is the compression ratio?

26. What are the disadvantages of supercharging a Diesel engine?

27. What problems are created by the use of supercharging with respect to cooling. Name some of the advantages?

28. What is the weight of the engine, in pounds per horsepower, for the General Motors engine? What type of blower does it employ? What is the brake horsepower per cubic inch of piston displacement?

29. What is the piston speed of the Fairbanks Morse, two-cycle, marine engine? What is the brake mean effective pressure at rated load? What is the displacement, per brake horsepower, in cubic inches?

30. What type of engine is the Hamilton, H.O.R. marine engine?

31. What is its mean effective pressure? What is the air-fuel ratio? How much excess air is this? (H.O.R.)

32. What is the piston displacement, brake horsepower per cubic inch of piston displacement, and weight per pound per brake horsepower? (H.O.R.)

33. What are the advantages of the gas-Diesel engine made by Cooper-Bessemer? How is the change-over from one fuel to the other accomplished?

34. What are some of the possible applications of this type of engine?

35. What are the possibilities of using a cheaper grade of fuel oil in such an engine?

36. Is better volumetric efficiency at higher rotative speeds made possible by supercharging?

37. On what type of installations are the electrically driven blowers or superchargers used?

38. What is the Roots-type blower?

39. What types of injector are used on the General Motors engines? The Fairbanks Morse engines? The Hamilton H.O.R. engine?

40. Does the direct seating of the valves in the cylinder head permit more effective cooling than the cage arrangement?

41. What is the area of the main bearing surface (square inches) of the Cummins 250-hp engine? What is the ratio to the area of the top of the piston with a diameter of 7 in.? ($\pi/4 \times D^2$ = area)

42. What is the advantage of using welded steel plate in the fabrication of frames of large engines?

43. Are heat stresses more severe in the supercharged engine?

44. How does the turboblower cause back pressure and does this contribute to exhaust valve troubles? Why?

CHAPTER 3

BASIC PROBLEMS OF OPERATION

PROPER and accurate analysis of the basic problems of Diesel engine operation is the first concern of the maintenance engineer. The higher cost of Diesel maintenance is due to the shorter life of liners and pistons, rings and bearings, and to accelerated wear of the crankshaft and other rotating parts. Just how these elements of the Diesel engine are being gradually and surely improved through a better knowledge of and experience in solving the problems involved will now be presented in more detail.

The natural handicaps of Diesel engine operation are being gradually overcome by the improvements and development of better working parts and more intelligent application of the Diesel engine. Practically every mechanical problem peculiar to the high-speed Diesel engine had to be worked out independently and means perfected to insure efficient performance. Although the Diesel and gasoline engine had much in common, gas engine experience afforded little comfort to the operator of the Diesel engine, since the operating principle is basically different.

In the present stage of the development of the high-speed Diesel engine, what operating-problems and design-difficulties have been solved, and what are some of the important problems remaining to be solved?

Any effort to supply the answer to this question would be far from satisfactory without looking to the engine builder and designer for much of the information and experience rapidly accumulating. In subsequent chapters, this book will undertake to describe the details of the maintenance and operating procedure which have been found resultful in the field.

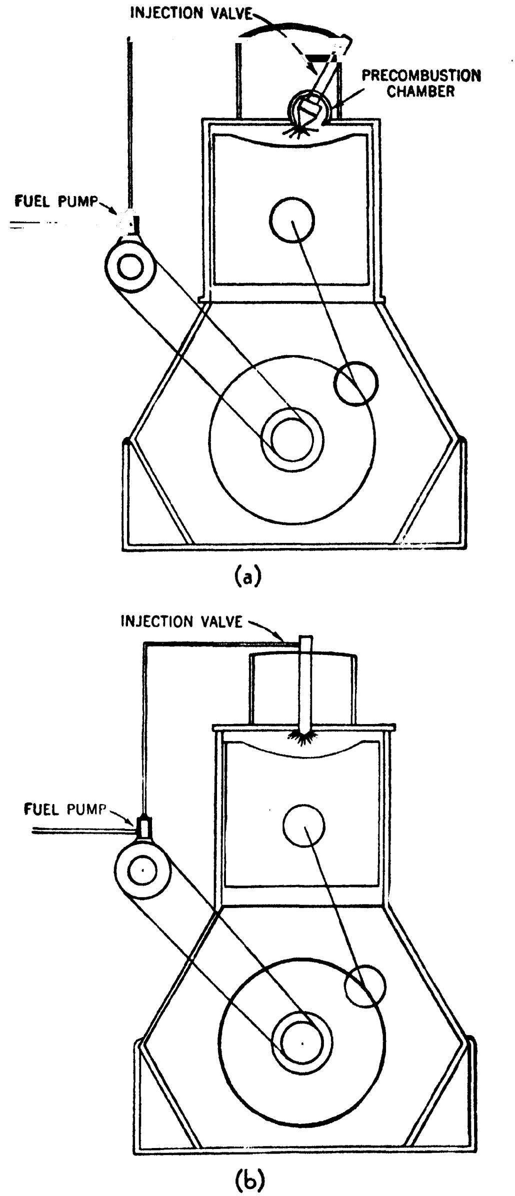


FIG. 3-1. (a) Typical precombustion chamber engine. (b) Typical open-combustion chamber engine.

Improvements of engine bearings. Formerly all engines used white-metal- or babbitt-lined bearings. The metal was spun into place or poured into the cast box or bearing cup, or it was made in the form of removable shells. High rotative speeds and high maximum pressures contributed to frequent bearing failures. The usual kind of failure started with cracking of the babbitt surface, thus weakening the metal. This weakening was followed by the spreading and pounding out of the metal, the final result being a complete failure of the bearing. Any effort to reduce this by better and improved lubrication was usually disappointing.

The fundamental weakness of this type of bearing was its low resistance to stress, low physical strength of the bearing metal under high operating temperatures and pressures. Copper-lead alloy afforded an approach to the solution of this problem. One engine used such an alloy made up of as much as 30 per cent lead and 70 per cent copper, but this proportion varied considerably with different bearings. When the mixture had a fine-grained homogeneous texture, it was more resistant to the heavy loads and high temperatures than any other material used up to that time. The use of this alloy was the first important development in bearings for the modern high-speed engine. A similar alloy is now widely used.

Bearings for the modern high-speed engine are known as the high-precision type and are made with such precision that they are neither bored nor reamed nor scraped to fit the journals when installed. Worn bearings are simply replaced with new shells and have proper precision fit unless the shaft is also worn beyond use or the clearance, as a result of wear, exceeds the allowable limit. Bearing shells are made of steel and the copper-lead alloy is applied to the shell. Some of these shells are only 0.075 in. thick and have bearing metal applied less than 0.025 in. in thickness.

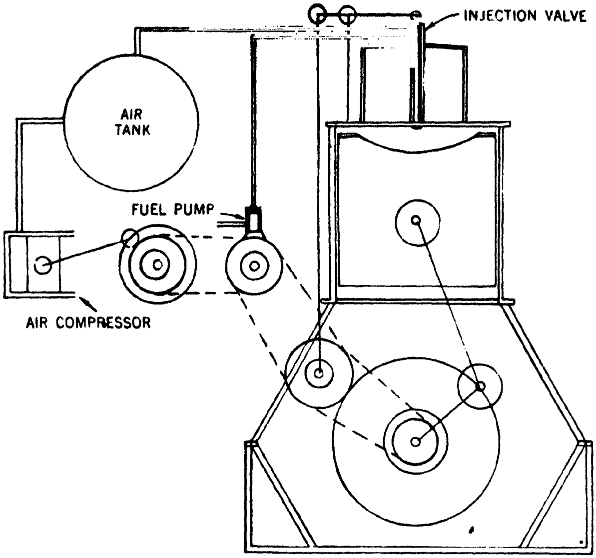
The connecting rod big-end bearings required special provisions, which consisted of machining the crankcase and details of the engine with great precision so that the interchangeability and replacement of the bearings was made practical. The final running clearance of the copper-lead alloy bearing is around 0.001 in. for each inch of shaft diameter. The crankshaft journals designed to use such bearings had to have a hardness greater than would have been needed for ordinary babbitted bearings. When a new set of replacement bearings is being

run in, care must be used to avoid cutting the shaft. The chief method of overcoming this difficulty is an ample supply of high-grade lubricating oil at all load points, usually a high viscosity oil being needed for running in the bearings. Keeping the oil clear and clean is a definite requirement also. After the running-in period of a few hours, when the surfaces of both the shaft and the bearing indicate a surface in perfect condition, the use of a lower viscosity lubricating oil is found to be safe and satisfactory.

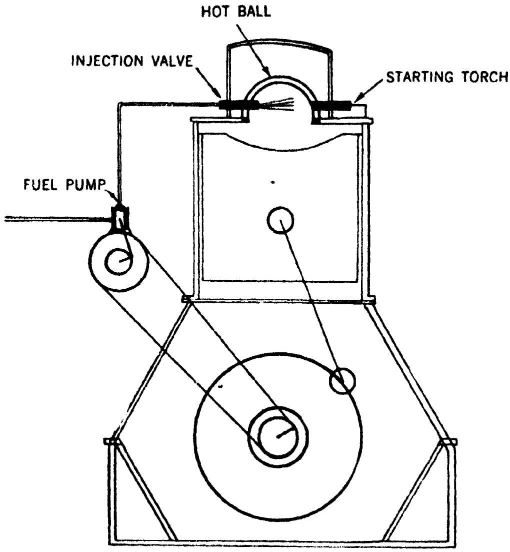
Cylinder liner problems. Cylinder liners were originally suggested to permit easy and low-cost replacement. In the early days of the automotive Diesel engine for trucks and tractors, cylinder wear was pronounced, and it was accepted as a necessary evil. This rapid wear of the cylinder liner was often due to faulty air and lubricating oil filters. Improper lubrication occasioned by rapid dilution of the lubricating oil in the crankcase was another common experience. The wear of the Diesel engine liner was much greater than that of the gasoline engine for the same service. While the wear in such engines was directly related to dirt drawn into the engine and dirty lubricating oil, to this had to be added the greater gas pressures produced in the cylinder, the higher temperatures of combustion, the residues of that combustion, dirty fuels, and the corrosive influence of condensation resulting from poor or incomplete combustion.

The problem of rapid wear engaged the Diesel engine builder for many years and is not altogether solved at the present time. As a result of years of effort by many manufacturers, the rate of wear of a Diesel cylinder liner in many present-day Diesel engines is actually lower than that found in competitive types of gasoline engines. The chief contribution to the solution of wear of the cylinder liner was the development of a cylinder liner material, a heat-treated, hardened, ground and honed nickel and chrome alloy cast-iron cylinder sleeve that stands up under the conditions of operation. Metallurgical difficulties were eliminated by trial and error with various metal compositions until now leading manufacturers can predict the wear of most compositions. Much research is still in progress on this subject and new developments are being reported frequently.

While some of these cylinders have a Brinnel of 450–550, the mere characteristic of hardness is not the principal preventive of engine wear. Grain structure and finish are also important in low-wear results.



(b)



(a)

FIG. 3-2. (a) Air ignition. (b) Surface ignition.

It was found that a fine-grained texture that would take a high, resistant polish is determined by the correct use of chromium, molybdenum, and nickel alloying elements coupled with a series of heat-treating processes carried on during the manufacture of the liner. Such liners naturally cost more than gray iron cast liners, but the longer useful life offsets the extra cost. Maintenance experience supports the conclusion that the overall cost of operation and maintenance with this high-grade sleeve is on a par with gasoline engines in similar services. The rate of wear of these liners is shown by experience to be less than 0.001 in. per 500 hr of operation, when the engine is operated with clear air, clean fuel, and a clean engine designed within the limits of the application. Additional satisfaction with the life of the liner is traced to the use of better lubrication for the piston rings and improvement of the piston rings both in design and material.

Heat problems in piston rings. Piston rings designed for use in the high-speed Diesel engine were found to present a different problem from those for the gasoline engine. Simply making the ring strong enough to stand up under the higher working pressures and shock forces common to the Diesel was not the answer to this problem. Special piston ring iron and structural characteristics were worked out. Much of the problem was related to the piston itself. The conventional cast-iron piston was not entirely satisfactory from an operating standpoint and was too heavy for rotative speeds as high as 2000 rpm. Vibrations were severe with heavy pistons. The cast-iron piston head did not stand up without scalding and cracking under the high-temperature and high-pressure blows of Diesel combustion. In heavy-duty engines, however, no such problems occurred.

Alloy pistons. Light-weight metal alloy pistons have a low expansion coefficient. When properly heat-treated to increase the strength, they prove satisfactory in Diesel engines of the high-speed design. The Diesel engine operates on a high temperature heat cycle and there are high mean temperatures in the piston itself. This high temperature causes a breakdown of the lubricating oil in service, as indicated by stuck rings and a high rate of liner wear. Much effort has been made to simplify and improve piston design in order to obtain a low enough temperature condition for the piston rings, to prevent carbon and gum formation, to make lubrication more effective, and

thus reduce liner wear and ring deterioration. Effective methods of keeping piston rings in good working order for long periods have been sought from many angles.

The particular system of combustion employed in the engine influences to no small degree the high temperature of the piston head and the resulting heat problems. This will be discussed later.

Piston ring improvements. Diesel piston rings must operate in a high temperature zone where the residues of combustion are coupled with inadequate lubrication and where the conditions are so severe that the top ring cannot stand up for more than a certain length of time. This first ring is subjected to high bearing pressures against the cylinder liner as a result of high gas pressure. The common gasoline engine piston ring is practically worthless in the Diesel engine. Piston rings necessarily have a shorter life than the liner has. One arrangement to relieve the ring of the burden of high temperature and pressure is to place the top ring lower or further from the head of the piston than is the practice for the gasoline engine.

Adequate wall pressure by the piston ring is required for sealing and this must be maintained for new rings as well as worn rings. The metallurgical advancement in the making of Diesel piston rings includes the electric furnaces for iron melting and refining and very accurate control. While piston rings are made of alloy cast iron, they are not heat-treated, for experience shows that the ring must be considerably softer than the liner, with the ring wear being several times that of the liner. On the other hand, when piston ring wear can be reduced, there is a parallel reduction in liner wear. The facts are that the useful life of piston rings is being increased by better rings and better maintenance and operation.

When hard rings are used with hard liners during the running-in period, some scuffing of the rings usually occur, a matter that affects both the life of the rings and the liner. The danger of this scuffing is eliminated when high-viscosity lubricating oil is used during the running-in period.

Development of combustion systems. There are two broad classes of combustion systems, the mixing-chamber type and the direct-injection type. These are represented by Fig. 3-1 and Fig. 3-2, respectively

1. Direct-injection system. The direct-injection system usually operates at a lower compression ratio than does the mixing-

chamber type. Such engines are easier to start and have a lower fuel consumption, but the combustion is somewhat rough and the engine consequently rough in operation. In order to avoid too rough running, this type of engine has been restricted to lower brake mean effective pressures. The injection nozzles used with this system necessarily have a number of very small orifices for spray atomization and for penetration.

Some of the maintenance troubles are the result of clogged orifices, resulting in poor atomization, which is accompanied by smoky exhaust. As the soot accumulates, the lubricating oil is diluted and the crankcase accumulates carbon deposits, which have washed down from the combustion spaces.

2. *Mixing chambers.* The mixing-chamber class of combustion systems consists of either the precombustion chamber arrangement or the spherical turbulent chamber. The volume of the precombustion chamber is usually around 35 per cent of the total clearance volume. Complete combustion at full load does not occur in the chamber. The fuel already burning is ejected through the throat into the main cylinder above the piston. The burning is controlled since a definite amount of time is involved in this combustion process, which occurs in two stages. One advantage is quiet operation through controlled combustion by means of the simple single, large-orifice injection nozzle. This type of nozzle is comparatively trouble-free. The fuel consumption of such engines is fairly economical and the power output is large. One disadvantage is that cold starting is a little more difficult than with the open chamber with direct injection. Such engines also have some tendency toward a smoky exhaust at heavy loads, especially when low grades of fuel are used.

The spherical turbulent chamber operates on the basis of air turbulence to improve mixing and burning. Its volume may range from 50 to 70 per cent of the total clearance volume. The nature of the combustion process is similar to that which occurs in the precombustion chamber design. This type of chamber has been highly developed and gives very low fuel consumption and very little smoke in the exhaust. When operating at high compression ratio, the turbulence chamber is satisfactory in starting down to freezing temperature. The combustion is somewhat rough and noisy, but considerable improvement has lately been made in this respect. Water-cooled injection valves, insulated pistons, and shielded com-

bustion chambers are some of the improvements undertaken with this class of engine. Compression ratios as high as 18 to 1 have been reported, and the development of the shape of the chamber to reduce surface to volume ratio has been undertaken, in order to obtain more complete combustion and cleaner exhaust without smoke.

Pressure lubrication. Several years of intensive development under actual field conditions have been required to refine and improve the details of pressure lubricating systems, so that they are dependable and reliable according to engine builders. One difficulty has always been that oil is so viscous in cold weather that the pressure in the circulating pump rises to a high point and overloads the oil pump drive. Some sort of relief valve at the pump, set to open when the pressure rises too high, has usually been adopted to avoid these high pressures. When large amounts of oil are circulated, all points are lubricated, and this lubrication helps cool the hot metal engine parts at high rotative speeds. Careful attention has been given to the matter of preventing the fine dirt and grit from finding its way into the oil and being carried to the bearings. External oil lines have almost been eliminated. The brazed tubing used to convey oil to the bearings has been eliminated in some engines, and in its place are drill passages in the pump body. Cored and drilled passages in the crankcase casting supply oil to the main bearings.

Proper lubricating oil pressure. It has been found that oil pressure must be kept as low as practical in order to reduce lubricating oil consumption and prevent oil pumping. Consequently the amount of oil circulated depends upon the pressure to be maintained in the system. Very few engines operate with lubricating oil pressures higher than 25 to 30 psi, a pressure that generally gives all-round good results, but some engines operate with a lower pressure. When filters become clogged and cut off the flow of oil to the bearings, serious results follow. This is prevented by the use of a by-pass valve between inlet and outlet of the filter to open at a set pressure. Sometimes these by-pass valves give trouble as a result of sticking, but some of the latest developments have overcome this difficulty. One of these valves was so designed that it was easily removable for inspection, and it was nonadjustable to prevent the troubles usually occurring with the adjustable type when faulty adjustment was made by inexperienced operators.

Valve stem lubrication. Much trouble has been experienced as a result of failure to keep the valve stems lubricated. The operator either did not oil valves or used excessive amounts under the average conditions. Practical schemes have been worked out to insure lubrication of the valve stems, such as a strip of felt resting on the rocker arm to prevent an oversupply. The amount of oil which flows is governed by the width of the felt strip in contact with the rocker arm.

Filter difficulties. Effectiveness of filtering depends upon the rate of circulation of the oil through the filter element. The metal filter is widely used and preferred from the standpoint of servicing and cleaning. However, this type of filter is not so efficient as a filter that has an area of approximately 100 in. per gallon per minute of oil circulation. This is equivalent to a volume of 4 cu in. per second passing through the filter. Failure to clean and service filters is more often the trouble than the filter itself.

Intake air filtering. The problem of filtering the air for a Diesel engine requires consideration of several factors. Since the Diesel engine operates at wide-open throttle conditions and low minimum depression, a larger filter than would be satisfactory on the gasoline engine is required. This ratio is about 5 to 3 compared with the gasoline engine. A filter that is too small is one of the first troubles to look for on a Diesel engine.

Efficiency of the filter is the next most important item since any silica dirt that passes the filter and is deposited on the cylinder walls increases the wear of the liner as well as the rings. The failure to keep air cleaners free of dirt and accumulations is a common failing of the operator. Although the filter cannot be 100 per cent efficient even with the best care, it must be serviced promptly as instructed by the engine manufacturer or disastrous results may be experienced.

Filters for air intake for Diesel engines have been highly developed, the latest types being nonclogging even in air carrying a large percentage of dirt and dust.

Importance of clean fuel oil. No requirement of operation is of greater importance than clean fuel oil, and the experience with this problem indicate the necessity of providing clean fuel at all times. Special equipment such as rustproof supply tanks used in direct connection with the engine are now specified. Experience has shown the need for cylindrical strainers fitted over the outlet in the bottom of the tank and centered by

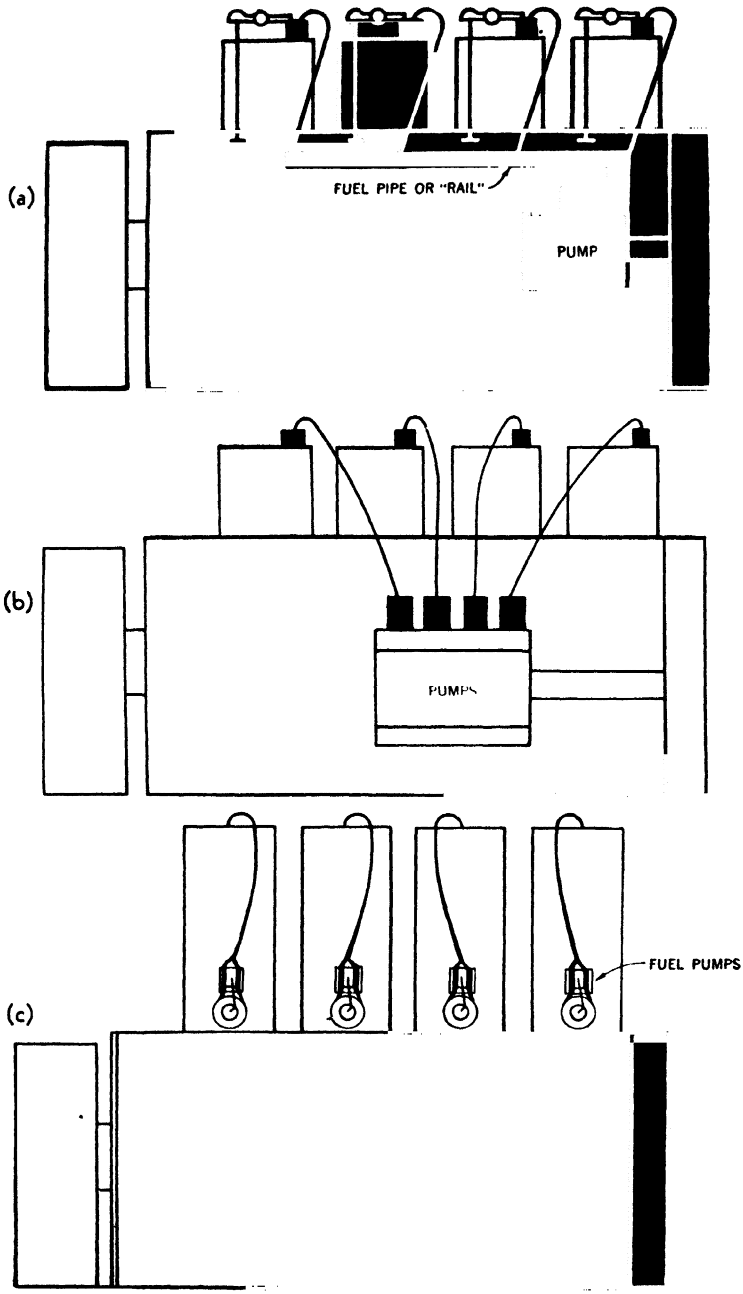


FIG. 3-3. (a) Common rail system. (b) Mechanical injection, individual pumps. (c) Individual pump arrangement.

the filter opening flange at the top. The strainer provides two passages of the fuel through successive stages of filtering. Storage tanks are also constructed to provide a strainer consisting of mesh bronze wire cage through which all fuel going to the tank passes. The bottom of such tanks has a large drain for water, and settlings are drained off per instructions.

Water traps and filters are also placed between the tank and the injection pump. When water is in a state of emulsion, there is great difficulty in extracting it from the fuel. The water trap works on the principle of intimate surface contact between the liquid and metal, and the difference in surface tension of a globule of water and fuel. The low velocity movement of the fuel stream also contributes to the effective operation of the water trap. For proper handling, the fuel is usually pumped against a low pressure of approximately 25 psi through a large area of virgin wool triple-weave cloth at the rate of 1000 sq in. of cloth to a gallon per minute. The fuel next passes through an edge filter, averaging 0.0015-in. mesh, to the injection pumps.

Sometimes an operator may permit dirt to get into the system when disassembling and inspecting the apparatus. The use of a jet-type filter at each suction port just above the suction valve has proved effective in excluding dirt from the pump barrel when it has been inadvertently left in the system. A jet filter of the same type, but usually with a 0.001-in. spacing, is used in connection with the injection nozzle, insuring effective operation of the nozzle without dirt and grit that might have been by-passed from the pump.

Complete rustproofing of all the parts of the fuel injection system and the use of hardened and corrosion-resisting alloy steel for moving parts in contact with the fuel are contributing factors to the more successful performance of Diesel injection systems on modern engines.

Lubricating oil coolers. While coolers or heat exchangers are not necessary in every case, they are highly beneficial because cool lubricating oil maintains its viscosity and thereby lubricates more efficiently than does hot oil. The engine builders have settled the question of how cool the oil should be kept for engines, and in most of the instruction books the instructions are very definite. The usual cooler has sufficient capacity to cool the oil to 135 to 165° F. The size of the cooler needed depends upon the cooling-water conditions; the climatic requirements influence the size to some extent.

Oil purifiers. In stationary power plants and on marine installations, the centrifugal oil purifier for cleaning the fuel oil has come into general use. Fuels are delivered under conditions which sometimes result in dirty fuels, which, of course, must be cleaned. The purifier removes from the fuel all water and sediment and some of the ash-forming constituents. Centrifugal purifiers greatly aid in reducing the rate of engine wear through cleaning the fuel before it enters the supply tanks.

Guides to Operation and Maintenance

A program for operators of Diesel engines that covers minimum requirements for success was outlined by the Field Service Department of Cooper-Bessemer Corporation, which stresses the importance of keeping a running account of the engine. The first and most important step is keeping a log sheet and record. This device furnishes data that will indicate when repairs are needed, and what precautionary measures are required for anticipating problems of basic operation and maintenance. A sample log sheet is shown in Fig. 3-4.

The basic problems of operation as defined in this chapter should receive the recognition that experience justifies. The operating techniques involved and the routine practices and procedures presented by this guide to operation apply the lessons of experience to the prevention of maintenance troubles and help to insure better operation of the engine. The procedure now presented is representative of the best operating practices. The author is indebted to Ralph Boyer, Cooper Bessemer Corporation, for his help in this connection and for the use of this outline of operating procedures. Taking the log sheet as an indication of the steps to be followed and the data to be recorded, it is found that the most important considerations are the lubricating oil, the fuel oil, cooling water, air-starting system, temperatures of the water and oil, pressures of the oil and water. Thus, the temperatures, pressures, exhaust colors, and so on, are definite indications of the operating conditions.

Lubricating oil consumption. Check the level of the oil every day. Add to the supply when necessary. Make a careful record of the amount of oil added. The lubricating oil consumption is figured in horsepower hours per gallon. When the engine is operated at rated load and speed, the horsepower rating on the engine name plate is used and is multiplied by

Name of Company or Owner ______________________

Location ______________________

Engine Serial No. Unit No. Date: Sheet No. ____

Time	Temperatures				Pressures			Exhaust Temperatures						Engine Speed (rpm)
	Lube Into Engine	Oil Out Engine	Fresh Into Engine	Water Out Engine	Lube Oil	Fuel Oil	Start. Air	Cyl. No. 1	Cyl. No. 2	Cyl. No. 3	Cyl. No. 4	Cyl. No. 5	Cyl. No. 6	

Daily Summary

Total Hours Operated ____

Lube Oil Added ____

Fresh Water Added ____

Fuel Oil Supply ____

Ave. Load ____

Lube Oil Consumption ____

Remarks ______________________

Signed ______________________

Operator

OPERATOR'S CHECK LIST OF DAILY ROUTINE

DAILY PROGRAM

Operation		Previous Record	Today
1	**Measure engine oil level**	____	____
2	**Check governor oil level**	____	____
3	**Oil exhaust valves**	____	____
4	**Oil valve tappets**	____	____
5	**Measure fuel oil supply**	____	____
6	**Measure fresh water level**	____	____
7	**Blow out air tanks**	____	____
8	**Turn handles on lube oil strainers**	____	____
9	**Wipe engine down**	____	____
10	**Water temperature**	____	____
11	**Exhaust temperatures**	____	____
12	**Exhaust color and load condition**	____	____
13	**Lubricating oil temperature**	____	____
14	**Lubricating oil pressure**	____	____
15	**Fuel oil pressure**	____	____

FIG. 3-4. Sample log sheet for Diesel engine plants.

the number of hours run since the last oil was added. This is divided by the number of gallons used.

Example:

The engine is rated at 500 hp at 400 rpm. Since the last oil was added to the system, the engine ran 30 hr at rated speed. Assume that 4 gal of oil were added to bring level up to normal. Then the oil consumption is:

$$\frac{500 \times 30}{4} \times \frac{15000}{4} = 3750 \text{ bhp-hr per gallon}$$

It is difficult to calculate the lubricating oil consumption by this method when the engine is operated at variable load and speed. Instead of the above the following method is then preferred: Assume that the engine is new or in good operating condition. An accurate record should be kept of the total hours operated and the amount of lubricating oil consumed. From this data, the average lubricating oil consumption can be determined. After the average lubricating oil consumption is established for the operating condition, it is then an easy matter to determine any variation or detect deviation from this average. Whenever the average oil consumption drops off, or the level of the oil increases, look for evidence of fuel oil leaks and water leaks into the lubricating oil. Should the oil consumption increase suddenly, look for a leak in the oil piping, engine base, or cooler. Regular inspection of these parts is also indicated. Under normal operating conditions the lubricating oil consumption should generally remain constant. When piston ring and cylinder liners become worn, the lubricating oil consumption increases. Usually this increase is gradual as the wear rate progresses beyond the allowable limit. This problem will be discussed in complete detail in the chapter on pistons and rings. When the wear exceeds the allowable limit, there will be blow-by of the compression by the piston rings, and as this continues the engine efficiency will fall off, as indicated by increased fuel consumption for the same power output. It is evident that when the lubricating oil consumption increases in a normal manner to this point, it is no longer economical to operate the engine in such a condition. A general overhaul is then in order and should include checking of cylinder liners for wear, replacement of ring, and installation of new pistons. Planning an overhaul and inspection of such parts will be presented in the chapters on maintenance procedures.

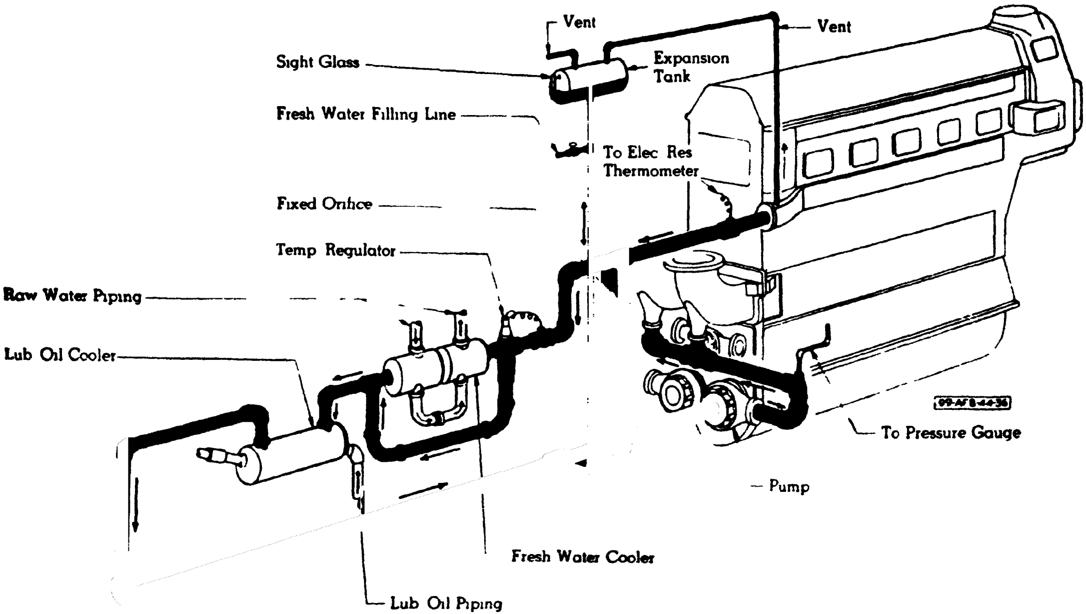


FIG. 3-5. Fairbanks Morse opposed-piston Diesel engine – layout of closed cooling system showing piping arrangement.

Special governor lubricating requirements. Additive or detergent oils, ordinarily used to lubricate the engine, should never be used in the governor. The oil level in the governor should be checked daily, and only clean oil added. When oil must be added to the governor more than once a week, the governor should be removed and the oil seal around the drive shaft replaced, according to the governor instruction book. The governor is a delicate and a very finely constructed mechanism that deserves special care and attention.

Exhaust valve oiling. The exhaust valves should be oiled daily. The use of one of the following types of oil is recommended:

1. Penetrating oil.
2. A 50-50 mixture of kerosene and lubricating oil.
3. A 50-50 mixture of fuel oil and lubricating oil.

The best practice is never to use plain, undiluted lubricating oil and certainly never grease. The high temperatures to which the valves are exposed bake the oil and cause the valves to stick in the bushings or guides. On top of the cylinder head cover on some engines are small oil cups to which should be added a few drops of oil daily to lubricate the valve tappets.

Fuel oil supply. The cost of operation is determined by keeping a record of the amount of fuel used. When the fuel oil supply is checked daily and the proper amount added, a record is made of this on the log sheet.

Fresh water. Abnormal loss of water is an indication of leaks from the cooling system. The fresh-water supply is checked and water added to the system when necessary. A record of water added in some installations should be kept. The importance of amount evaporated or lost depends upon the design of the cooling system. A routine check should be made regularly of the water pump packing, piping connections to the engine jackets, such as the water jumpers, and the liner seals for leaks of all kinds. When excessive leaks are found or when water gets into the lubricating oil system, the engine should be shut down and the leaks repaired. When the leaks are small, the fact can be indicated on the job chart and the leaks repaired at the next overhaul or maintenance period.

Drain air tanks. Once a day, the blow-out valve in the bottom of the air tanks should be opened and any accumulation of moisture, oil, or dirt blown out. Failure to do this permits the accumulations to contribute to starting difficulties.

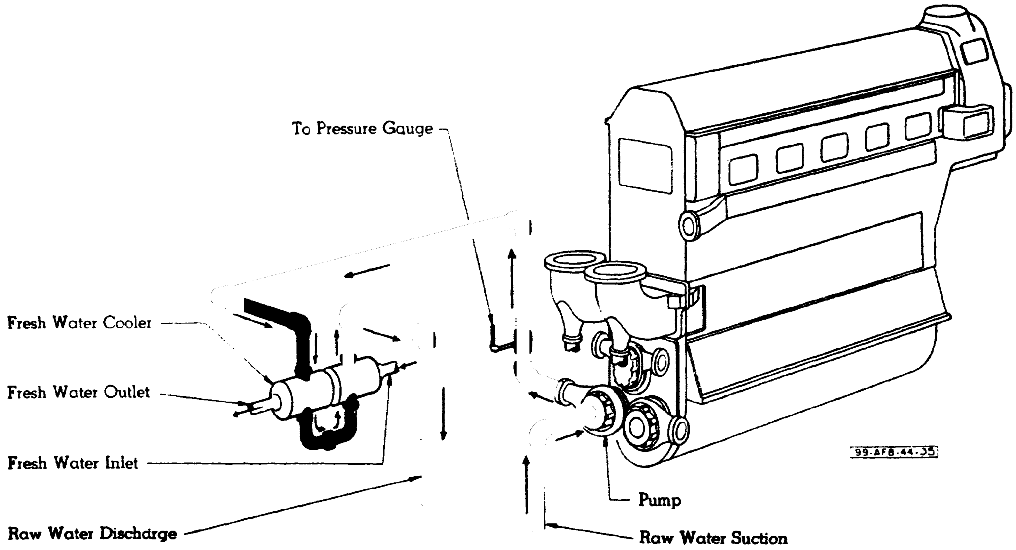


FIG. 3-6. Fairbanks Morse opposed-piston engine—layout of open, or raw-water, cooling system.

Lubricating oil strainers. The handles of the lubricating oil strainers should be turned several times a day. There is no harm in turning them often; they are less liable to clog when they are turned regularly. However, never force the handle. If it is difficult to turn, find out why.

Care of engine. A good operator keeps his engine clean. Dirt around an engine is harmful. Cleanliness pays good dividends. Whenever water and oil leaks are found, they should be fixed if they are serious, and if small, a record should be made in the repair job book so that the work can be done at the earliest maintenance period. At the same time, find any loose bolts and nuts, and tighten them. Loose nuts and bolts indicate loose operation.

Temperature records. The importance of temperature observations and recording cannot be overestimated. Good thermometers and pyrometers, kept in proper operation condition, are essential. The following temperatures should be given careful attention.

Water temperatures. The temperature of the water in the cooling system should be observed and recorded in the engine log book at regular intervals. Correct water temperature is an indication that the cooling system is functioning properly. When the temperature is above normal, the trouble should be investigated at once, for it is trouble. The following are steps in checking the cooling system:

1. Adjustment of the by-pass valve at the heat exchanger or valve that is not properly set somewhere in the piping.
2. Water supply—see that there is plenty in the system.
3. Lack of circulation—this may be caused by failure of the pumps, a plugged heat exchanger, or radiator, oil cooling, or stoppage in the piping.
4. Restriction in the engine water-jacket cooling spaces or passages in the water circuits.

When the trouble is not located almost immediately, shut down the engine and investigate the water system thoroughly. Every operator must be thoroughly familiar with the details of the cooling system used on his engine. Prints of the original piping drawings should be obtained from the manufacturer of the engine.

When the water temperature is below normal, the by-pass valve adjustment at the heat exchanger should be checked. There is an automatic three-way valve on some engines located

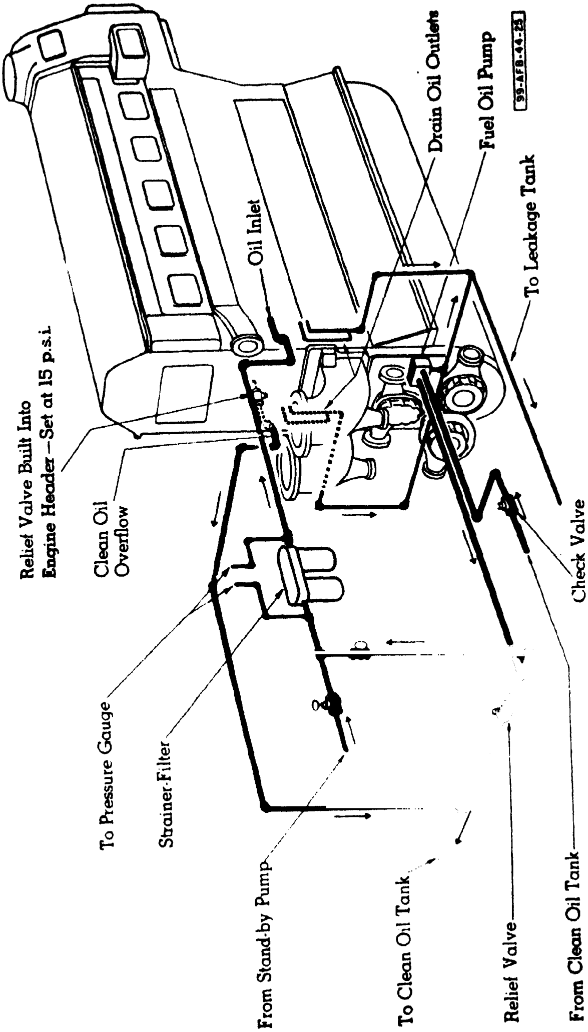
at the heat exchanger. A routine program should be set up to inspect this valve periodically to keep it in working order at all times. Likewise, some engines are equipped with an automatic water temperature alarm or a shutdown device. Unless this device is kept in proper working order by regular inspection, it cannot be depended upon to do its duty. After the engine is started, the by-pass valve at the heat exchanger should be adjusted to bring the water temperature to normal for the temperature of the weather.

Exhaust temperatures. It is an established practice to observe and record exhaust temperatures on the engine log sheet several times daily. The temperatures of each cylinder are recorded at the same time. Unbalanced temperatures between any two cylinders are watched carefully. The exhaust temperature between any two cylinders should never vary more than 75° F at full or three-fourths load. When any unbalance is evident, inspection should be made and the following adjustments made in order of their importance:

1. Inspect pyrometer to make sure it is in good working order. It may be necessary to remove and inspect the thermocouples; the wire and connections should also be inspected.
2. Remove the inspection nozzle from the cylinder whose exhaust temperature is out of line and place it in the nozzle holder. The removed nozzle should then be tested with the priming pump, making certain that the initial pressure setting is correct and that the spray pattern is also correct. If the nozzle is found to need adjustment, remove all nozzles on the engine and reset them in accordance with the engine instruction book, in which the proper procedure for nozzle adjustment is found.
3. If the nozzle is found to be in proper working order, adjust the fuel injector tappets for the proper *duration* and *lift* and check the fuel injection timing. The instruction book details the proper procedure for this also. The operator must know the correct timing of the engine, the speed, and the type of service. When he is unable to determine this for himself, he consults the field service department of the manufacturer.

When all temperatures of the exhaust are higher than the average temperature at the normal load and speed, it may be an indication that the engine is overloaded or that the main valve timing is incorrect.

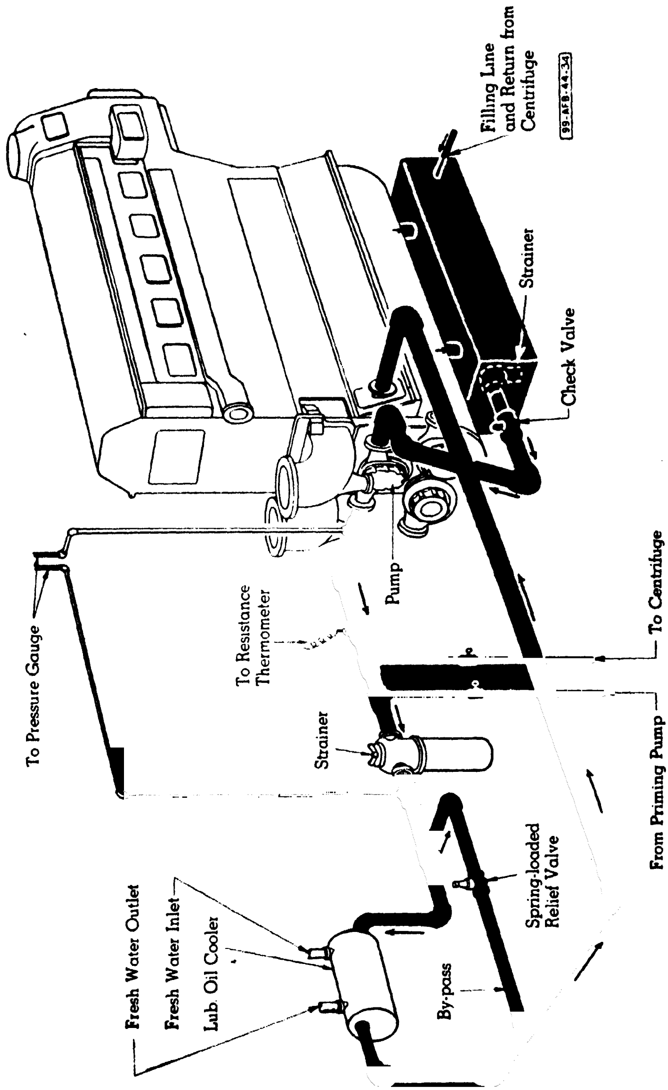
Exhaust color and load condition. When the exhaust pipe emits the least amount of smoke, this is an indication that the



Relief Valve Built Into
Engine Header—Set at 15 p.s.i.
Clean Oil
Overflow
Oil Inlet
Drain Oil Outlets
Fuel Oil Pump
99-AFB-44-25
To Leakage Tank
Check Valve
To Pressure Gauge
Strainer-Filter
From Stand-by Pump
To Clean Oil Tank
Relief Valve
From Clean Oil Tank

FIG. 3-7. Fuel oil piping—Fairbanks Morse opposed-piston engine. The fuel system consists of the supply and injection systems. The supply system includes the clean oil service tank, stand-by pump, built-in supply pump on the engine, strainer-filter, gauges, and the necessary piping and fittings. The fuel supply pump is driven by gears from the lower crankshaft. The supply pump draws fuel from the clean oil service tank and delivers it through the strainer-filter to the engine inlet. The capacity of this pump is such that more fuel is pumped into the header than is needed by the injection pumps with the result that a pressure of about 15 lb is built up and maintained in the header. A relief valve built into the header regulates the pressure, allowing the excess fuel to return to the tank.

The fuel oil header consists of a pipe which is connected to each injection pump. Fuel from the strainer-filter is circulated through the headers and returned to the service tank by the supply pump. Fuel required enters the injection pump, and the excess returns to the service tank through the return pipe on the outside of the opposite control side of the engine. The capacity of the supply pump is such that a sufficient velocity is obtained in the injection pump headers to insure rapid replacement of fuel as it is used by the individual injection pumps.



Filling Line
and Return from
Centrifuge
99-AFB-44-34
Strainer
Check Valve
To Centrifuge
Pump
To Resistance
Thermometer
To Pressure Gauge
Strainer
From Priming Pump
Spring-loaded
Relief Valve
By-pass
Lub. Oil Cooler
Fresh Water Inlet
Fresh Water Outlet

FIG. 8-8. Lubricating oil system outside the engine—Fairbanks Morse opposed-piston engine, showing piping and filters. This engine is equipped with a pressure lubrication and piston cooling system which supplies a continuous flow of oil to all surfaces requiring lubrication, and to the pistons for cooling. After engine lubrication and piston cooling, the excess oil drains to the oil pan and thence to the sump tank below the engine. The pressure system consists of a positive pressure gear pump and two oil headers built into the cylinder block, an oil sump tank, a strainer, coolers, thermometers, and the necessary piping and fittings, all of which may vary depending upon the application. The lubricating oil pump is mounted on the pump mounting plate on the control end of the engine, and is driven, through gears and a flexible coupling, by the lower crankshaft. The pump draws oil from the sump below the engine and forces it successively through the strainer and the coolers. In the coolers, the oil passes through tubes mounted in a core. Fresh water flows through the core cooling the oil. Leaving the cooler, the oil is piped to the engine, where it enters the lower lubricating oil header through an inlet flange. A spring-loaded relief valve is installed across one of the coolers to open at 15 lb pressure to insure circulation if the oil is congealed in the cooler when starting. In the line between the pump and the strainer is the bulb of an electrical resistance thermometer, the indicator for which is on the instrument board. From the same line, between the strainer and the coolers, a small pipe leads to a duplex pressure gauge on the instrument board.

engine is running under ideal conditions. Improper combustion and excessive oil burning account for the presence of smoke in the exhaust. The color of the exhaust should be observed at least daily. When the exhaust becomes smoky suddenly, the trouble should be investigated by checking the following:

1. The fuel injection nozzles.
2. Worn or sticking valves.
3. Overloaded engine.
4. Stuck piston rings.

Whenever the engine smokes at light loads and speeds, it is an indication that a high lubricating oil consumption has set in, and causes for this should then be investigated.

Lubricating oil temperatures. The lubricating oil temperatures are observed and recorded at regular intervals. The correct oil temperature is an indication that the lubricating system is functioning properly. When the temperature of the oil is abnormal or above normal, this trouble should be investigated at once by checking the following:

1. Setting of by-pass valves in the raw water at the oil cooler.
2. Oil pressure—check in accordance with oil pressure chart.
3. Oil supply.
4. Relief valves in the system—particularly at oil cooler.
5 Lack of oil circulation, due to failing pumps, plugged oil cooler, radiator, filters, strainers, or piping.
6 Restriction in engine oil passages, or any place in the circulating circuit.

Whenever the trouble is not located at once, shut down the engine and investigate the system thoroughly. Every operator must be thoroughly familiar with the lubricating oil system of the engine, and if not, the information in the instruction book must be studied. Severe damage may result when the engine is operated with abnormally high lubricating oil temperatures, which are always dangerous and indicate need of immediate correction. Failure to correct them is negligence on the part of the operator.

When the lubricating oil temperature is below normal, regulate the by-pass valves at the oil cooler. On some engines there is an automatic three-way valve at the heat exchanger, which calls for a program to be set up for this valve to keep it in proper working order. When engines are operated with the

lubricating oil temperature too low, the oil has a tendency to form sludge in the crankcase, with accompanying poor lubrication. Filter maintenance is increased by sludge of this kind.

After starting the engine it is necessary to adjust the by-pass valve at the oil cooler to bring the temperature to normal for the condition under which the engine is operating. It is also good practice to maintain the lubricating oil temperature at or near, but never exceeding, the cooling water temperature—a good rule to remember. Consult the instruction book, however, for all temperature requirements.

Engine lubricating oil pressures. Incorrect lubricating oil pressures definitely indicate one of the following:

1. A broken gauge or oil line.
2. Worn oil pump.
3. Leaking oil lines.
4. Relief valve out of adjustment.
5. Low oil level.
6. Sludge, or obstruction, in outlet from engine base or sump tank.

If pressures cannot be obtained after oil has reached operating temperatures, perhaps none of the above apply. If this is the case, to check for bearing clearances is next in order. Too much clearance indicates bearing wear.

Fuel pressures. Incorrect fuel oil pressure is usually an indication of several minor troubles:

1. A broken gauge or fuel line.
2. Failure of fuel pump.
3. Open nozzle, indicated by black exhaust.
4. Worn or improperly adjusted linkage.

Should the trouble be traced to the fuel pump, and should the fuel pressure go to zero as soon as the engine is stopped, check the valves in the fuel pump and distributor blocks for leakage, in accordance with the details outlined in the instruction book.

Monthly preventive maintenance schedule. Once every 600 hours, or at least once a month, the following monthly maintenance and inspection program should be carried out faithfully and in detail. Proper records should be made.

Lubricating oil strainers. Drain and remove the case on the lubricating oil strainers and clean it thoroughly before replacing. Look for evidence of babbitt particles in the lubricating oil

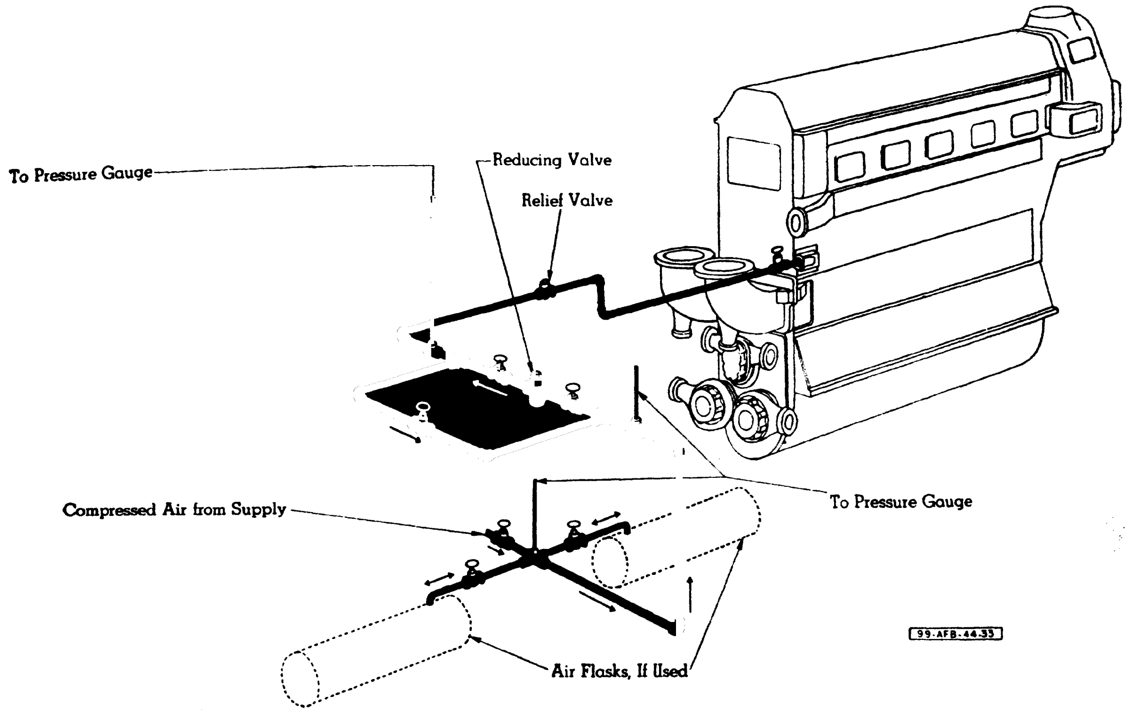


FIG. 3-9. Air starting piping, Fairbanks Morse opposed-piston engine.

strainer as this is an indication of bearing troubles or failures. Whenever babbitt is found, immediately examine main and connecting rod bearings. Find out the failure or start of failure if all must be removed. Details of bearing maintenance and inspection are given in subsequent chapters. Every effort to keep strainers clean is justified.

Lubricating oil filters. Clean the lubricating oil filters once a month, and oftener if necessary, as indicated by the condition of the oil. Some filters can be cleaned while the engine is in operation, whereas others require that the engine be stopped. The instructions on the filter give information on its care.

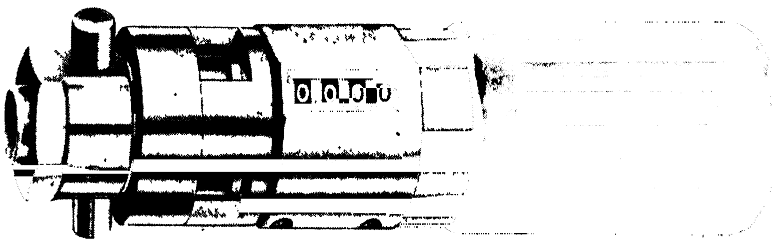


FIG 3-10. Bacharach Diesel pressure indicator for direct reading of compression and firing pressures of any type of Diesel engine at any speed or rate of pressure rise.

After operating the engine for a few months it may be possible to deviate from the monthly schedule for cleaning the filters, as the operating conditions vary from one installation to another and from month to month. This makes it difficult to set up a definite time schedule for this matter of cleaning filters. Experience, however, will show the need for the work.

Grease fittings. Fill all grease fittings on the engine once a month, using proper grease, and being careful to use ball-bearing grease for ball bearings. Have a list of proper greases for the engine and use them.

Overspeed device. An overspeed shutdown device is used on the engine, and this should be checked once a month to make sure that it is in proper operating condition. Usually the overspeed is set about 10 per cent above normal rated speed. Test by overspeeding the engine until the overspeed trips at the proper speed.

Day-tank inspection. Open the drain valve at the bottom of the fuel oil day tank and catch the drain-off in a bucket or can. Make every effort to get all the dirt and condensate out of the tank. Flush and clean if necessary. The water should be drained from the bottom of the storage or supply tank once a month, and the lines checked for rust accumulations, sediment, and foreign matter in general.

Top overhaul. Remove the cylinder head covers at least once a month and inspect the general condition of the main valves, rocker arms, and so on. Look for unusual oil streams, fuel oil leakage, blow-by, water leaks, and the like. When troubles are found, they must be corrected as soon as possible, and a record made of the findings so that an overhaul can be scheduled. Check tappets for clearance and all valves at the rocker arms. Consult the instruction book for details of top overhaul.

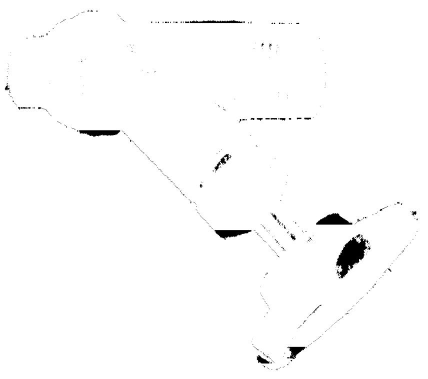


FIG. 3-11. Adapter for connecting indicator to cylinder cocks.

Compression and firing pressures. The peak pressures should be taken once a month if possible. The proper performance and engine balance is indicated by these pressures. The variation in pressures between cylinders should not exceed a maximum of 100 psi. If pressures increase or decrease over the last reading, the reasons should be investigated and the observations from the log sheet studied to determine the underlying conditions that may contribute to the unbalance. Peak pressures are affected by nozzle condition, fuel timing, and load on the engine. Ring conditions and linear wear may also be related, as well as leaking valves and ruptured gaskets.

While checking the peak or firing pressures check the compression pressures. To do this close the shutoff valve at the fuel injector for the cylinder whose pressure is to be measured. Compression pressures are influenced by piston ring condition and liner wear as well as leaky valve conditions. Whenever

compression pressures fall below 375 psi, the engine is due for an overhaul and reconditioning.

At six-month intervals, the main engine valve timing and the like should be checked. This also applies to checking timing chains, valve tappet clearances, and the lubricating oil drive shafts for pumps, and to lubricating the coupling.

Relation of cylinder pressures and exhaust temperature. The following is a guide to using the pressure indicator and the pyrometer:

High Compression Pressures

1. With low exhaust temperature
   a. Cylinder gasket too *thin*
   b. Shim between connecting rod and crankbox too *thick*

Low Compression Pressures

1. With high exhaust temperature
   a. Leaky valves
   b. Leaky or stuck piston rings
   c. Worn cylinder liner
2. With low exhaust temperature
   a. Cylinder gasket too thick
   b. Shim between connecting rod and crankpin too *thin*
   c. Excessive bearing clearance
3. With normal exhaust temperature
   a. Air cleaner or silencer clogged
   b. Blower air delivery below par

High Firing Pressures

1. With high exhaust temperature
   a. Fuel rack too far in
2. With low exhaust temperature
   a. Injection timing early
3. With exhaust temperature normal
   a. Eroded orifices of injection nozzle

Low Firing Pressures

1. With high exhaust temperature
   a. Injection timing late
   b. Exhaust valve clearance too small
   c. Injection nozzle dirty or leaking

2. With low exhaust temperature
   a. Fuel rack too far out
3. With normal exhaust temperature
   a. Low cetane fuel
   b. Air cleaner or silencer clogged

Normal Firing Pressures

1. With high exhaust temperature
   a. Overload
   b. High exhaust back pressure
2. With low exhaust temperature
   a. Light load
   b. Thermocouples defective
3. With normal exhaust temperature
   a. No evidence of difficulty
   b. Normal operation, engine functioning satisfactorily

TROUBLE SHOOTER'S GUIDE

Pyrometer Readings	Pressure Indicator Readings					
	Compression Pressures			Firing Pressures		
Exhaust Temperatures	High	Low	Normal	High	Low	Normal
	HC	LC	NC	HF	LF	NF
High—HE	1	17, 18, 19	4	10	7, 8, 9	4, 5
Low—LE	2, 3	20, 21, 22	0	15	13	6, 4
Normal—NE	1	12, 23	1	13, 14	11, 12	1

Causes, Remedies:

1. Normal operation
2. Cylinder gasket too thin
3. Shim between connecting rod and crankpin box too thick
4. Overload
5. High exhaust back pressure
6. Light load
7. Injection timing late
8. Exhaust valve clearance too small
9. Injector nozzle dirty or leaking
10. Fuel rack too far in
11. Low cetane fuel
12. Air cleaner and silencer clogged

13. Fuel rack too far out
14. Dirty fuel
15. Injection timing early
16. Eroded orifices of injection nozzle
17. Leaky valves
18. Leaky or stuck rings
19. Worn cylinder liners
20. Cylinder gaskets too thick
21. Shims between connecting rod and crankpin box too thin
22. Excessive bearing clearance
23. Blower air delivery below par

Operator's Report—Engine No.______

Date______	Pyrometer Reading			Adjustments Made
Cylinder No.	NE	LE	HE	Inspected by______
1				
2				
3		LF*		
4				
5				
6				

* Example: LF—low firing with Cyl. #3. Low exhaust temperature. Cause—fuel rack too far out—13.

QUESTIONS

1. What was the fundamental weakness of the early high-speed engine bearings?

2. What is a precision bearing—and does it have to be fitted by hand when installed?

3. Why were cylinder liners originally used in engines?

4. What factors contributed to liner wear?

5. What were some of the early metallurgical problems with liners?

6. What were some of the heat problems in piston rings?

7. What is the source of heat problems in piston rings?

8. What improvements had to be made in piston rings for high-speed Diesel engines?

9. What two types of combustion chambers were developed early in the evolution of the high-speed Diesel engine?

10. Name the functions of the mixing systems for high-speed engines.

11. Why was pressure lubrication necessary, and what special feature of this system requires attention?

12. How is proper lubricating oil pressure determined?

13. What were some of the problems of valve stem lubrication?

14. What type of lubricating oil filter is preferred from the standpoint of servicing?

15. What service does an air filter require?

16. What is some of the foreign matter that gets into the lubricating oil supply, and how is this foreign matter to be kept out?

17. Name some of the filters used for fuel oil filtering.

18. Describe the lubricating oil cooler.

19. What is the proper lubricating oil temperature for high-speed engines?

20. What mechanical means is used to clean fuel oil and lubricating oil?

21. Why is it so essential to keep an accurate log sheet of daily operation?

22. How is the average lubricating oil consumption rate to be established?

23. What causes an increase in lubricating oil consumption?

24. What kind of oil should be used in the governor, and what kind should not be used?

25. Why is it not good practice to use plain undiluted lubricating oil on the exhaust valve?

26. What checks should be made on the fresh-water cooling system?

27. What care should be given to the filters and screens?

28. What is the importance of exhaust temperatures?

29. What adjustments should be made to correct unbalanced exhaust temperatures?

30. What load conditions may be indicated by exhaust color?

31. How often should compression and firing pressures be taken with the indicator?

32. What is the maximum difference in pressures between cylinders that should be permitted?

33. How does the pressure indicator help to determine when an overhaul of the engine is necessary?

34. What fuel injection difficulties will be diagnosed with the use of the pressure indicator?

35. How is the pressure indicator used?

36. Why is it important for the operator to have piping prints showing the layout of the piping system for the fuel, lubricating oil, fuel piping, air starting, and so on.

37. What may be the cause when the average exhaust temperature is higher than normal?

38. When a cylinder has lower or higher exhaust temperature, what checks should be made of the fuel injection nozzles?

39. What check should be made of the timing of the fuel injection?

40. Does the main valve timing sometimes affect the exhaust temperature?

41. Is it also important to check initial pressure setting of the fuel injection nozzles?

42. What may happen to the pyrometer thermocouples that will give a false reading of engine exhaust temperatures? What should be done?

43. If the engine is operated with lubricating oil temperatures too low, what will be the result?

44. Why should the lubricating oil temperature never exceed the temperature of the cooling water, generally speaking?

45. What is the operator's most reliable source of information on the operation of his engine—particularly, such matters as the temperatures of the water, oil, exhaust, and so on?

PROBLEMS

1. You observe black smoke in the exhaust of a 8-cylinder, 2-stroke cycle engine. The exhaust pyrometer, compression pressure, and firing pressure readings are:

Readings	Cylinder Number							
	1	2	3	4	5	6	7	8
Exhaust Temp. ° F.	900	925	910	895	900	905	900	900
Comp. Pressure psi	490	505	500	485	495	500	500	500
Firing Pressure psi	790	810	805	780	800	805	800	800

What would be the probable cause of the smoke?

2. The exhaust pyrometer, compression pressure, and firing pressure readings of an 8-cylinder, 4-stroke cycle engine are:

Readings	Cylinder Number							
	1	2	3	4	5	6	7	8
Exhaust Temp. ° F.	750	300	730	820	870	850	830	675
Comp. Pressure psi.	460	450	460	350	500	450	460	450
Firing Pressure psi.	800	450	900	625	910	810	600	800

You know from experience that 760° F exhaust temperature, 475 psi compression pressure and 780 psi firing pressure are normal for that engine. Check the probable defects or causes:

Defect or Causes	Cylinder Number							
	1	2	3	4	5	6	7	8
Too early injection								
Too late injection								
Fuel rack too far in								
Fuel rack too far out								
No ignition								
Dirty nozzle								
Leaking nozzle								
Leaking valves								
Ring blow-by								
Obstruction in air intake								
Fuel-cetane—viscosity, etc (See Chapter 12)								

a. If you suspect a nozzle to be defective, how would you check this to make sure?

b. You hear combustion knock in one cylinder. Its exhaust temperature is below normal. What is the probable trouble and what would you do about it?

c. You suspect that one cylinder is missing, but you have no exhaust pyrometer. How would you locate the faulty cylinder (Chapter 12)?

CHAPTER 4

METALLURGICAL PROBLEMS

Construction and Design of Engines

Materials used in manufacture of engines. The metallurgical problem in the Diesel engine is selecting the most suitable material, properly conditioned, for each particular application. No one metal will function satisfactorily in all possible applications of an engine. Some metals, with slight modifications, serve a number of purposes, such as cast iron when used in the frame, the cylinders, heads, liners, and pistons. Others, such as the white metals in the main bearings, are largely single-purpose metals.

Intensive effort on the part of the material manufacturers to improve their products has been going on for years, with the result that simple metals and alloys have been augmented by a number of complex alloys. This applies to the steels as well as to the aluminums, cast iron, and the bronzes. Many of these alloys have proved their worth, and are now recognized as essential, while others are still more or less on trial.

At a meeting of the Oil and Gas Power Division, ASME, at Pennsylvania State College, R. J. Allen, research engineer, discussed the metallurgical problems of the Diesel engine in some detail, and gave the benefits of long experience along this line by the engine manufacturers. Although, in his opinion, all the numerous alloys offered may serve useful purposes, it must be recognized that in many cases the difference is slight. Hence, generally speaking, the manufacturer of Diesel engines is better off by developing a single type to the maximum, rather than half trying out every new alloy brought out. The improvement in some particular characteristic of the metal due to a special alloy may be offset by some detrimental feature not shown until the parts are in service. Allen's further observations are worth

considering here in connection with the problem of engine maintenance.

Selection of materials. The physical characteristics of the more standard alloys are now well established. Charts are available showing the tensile properties and hardness numerals for many forms of treatment; and the corresponding structural conditions have been recorded by the microscope. For various

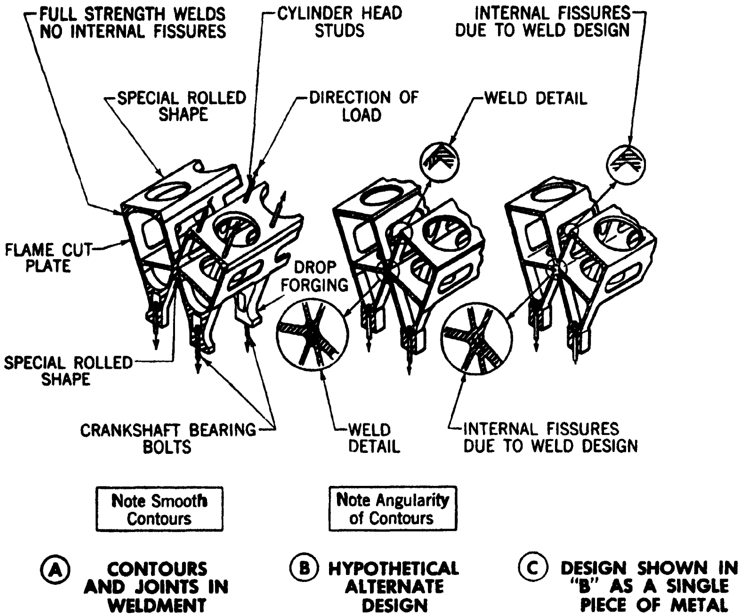


Fig. 4-1. Welded steel frame of a submarine Diesel engine. During 10,000 hr of operation at sea, the submarine was depth-charged several times. Because of vibrations, the engine frame suffered 126 separate fractures and cracks, which occurred during combat and were observed while the ship was at sea. Reported in *Mechanical Engineering*, February, 1945.

alloys the *endurance limits* are known under normal conditions, at elevated temperatures, and where *corrosion* influences are present. *Impact figures* are available for both the single-blow Izod and Charpy (Appendix I, "Metallurgical Glossary") and the repeated-blow Stanton types. In addition to this information, improved tests may be made, approximating more nearly the actual working conditions than those obtained in any of the standards tests available, and the actual data may be developed concerning the probable performance of the material

when put into service. Hence, there is no reason for stressing material beyond its safe working limit under any condition of service.

To obtain the most from materials, they are suitably processed in all stages of manufacture. For a steel casting, for example, the conditions of manufacture must be such that the material is free from *blowholes*, *shrinks*, and *segregation*. Then it is heat-treated to relieve strains and to develop suitable homogeneous structure. Slighting any operation may ruin an otherwise excellent product. Materials are therefore selected on the basis of quality and from reputable sources.

When a type of material is selected for use because of certain desirable characteristics, such facts should form the basis of a specification. Such specification should be a concise statement of the characteristics and condition of the material which have been found most satisfactory. The function of a specification, like that of a blueprint drawing, is to present the shop or the supplier a clear picture of exactly what is wanted. However, neither is an assurance of quality, for quality can be determined only by adequate inspection.

Whenever possible, standard specifications of the national engineering societies are used. A standard material regularly made is likely to be better than a special material infrequently made. Furthermore, there is usually a range of selection in standard materials not available in special varieties. However, if the standard specifications do not merit the use of it, or meet the particular needs, such alterations in the specifications as seem desirable are made.

Inspection and testing of materials. Inspection of materials is more difficult than that of mechanical parts. Inspection of the parts is largely a matter of checking size, the correctness of which can be determined to any degree of accuracy desired. In checking materials, however, many factors are involved, some which are tangible, whereas others are rather intangible.

The checking of materials is handicapped by the destructive nature of most tests. Tests such as the X ray, magnetic, and hardness are made on finished pieces intended for production, but others such as those for tensile strength, impact, and fatigue (those most commonly used) must be made on representative pieces.

Hardness tests are simple to make and positive in result. They give considerable useful information as to the uniformity

of the product and are used frequently in the inspection of the finished product. The X ray, although an excellent tool for development purpose and for checking of special, highly stressed parts, cannot as yet be thought of as routine test equipment. The magnetic test may be used for checking symmetrical sections, but it is not of great value as a general test. Iron filings are extremely helpful in detecting cracks in a hardened piece of steel if the latter is suitably magnetized.

Acid etching or picking is used to facilitate the inspection of individual pieces for surface detects without affecting the utility of the product, and incidentally it can also be used to an excellent advantage in checking cross sections of the material for possible segregation.

The testing of large forgings and castings is facilitated through the use of test prolongations forged or cast integral with each piece. These prolongations are made sufficiently large for actual physical test purposes, or they may simply provide a means of obtaining a fractured surface for visual examination. With little experience, one can tell rather accurately, from the appearance of a fracture, the physical condition of the material.

General types of material used in Diesel engines. Several general types of material are used in the manufacture of engines.

Cast iron. Cast iron has for years been the basic material in engine construction, largely because of its low cost and the ease with which it can be cast into intricate shapes. One does not have to look back very far to the time when it was simply a remelted pig iron, modified perhaps as to silicon content, to obtain reasonable casting conditions. Eventually the introduction of steel scrap, in small percentages, gave the cupola operator an opportunity to control the carbon content to some extent and to produce a better graphite formation with corresponding increase in strength. From this beginning, high-strength irons were soon developed in which the entire charge consisted of steel scrap. The term "semisteel" now has little significance, since varying amounts of steel scrap are used in practically all cast iron made today.

Strength of cast iron. The strength of any given piece of cast iron is a function of its section, and this fact must be taken into consideration when making comparisons. When cast in a standard test bar section, the high-strength irons will show tensile figures in excess of 50,000 psi. With the introduction

of nickel, the strength may be increased to over 60,000 psi and machinability may be reasonably maintained. All cast irons, unfortunately, are inherently brittle, and this fact must be taken into consideration when evaluating the usefulness of the high strength.

In plain cast iron the hardness is largely a function of the thickness of the section and the silicon content. If the casting is rather irregular in shape, the heavy sections may be soft

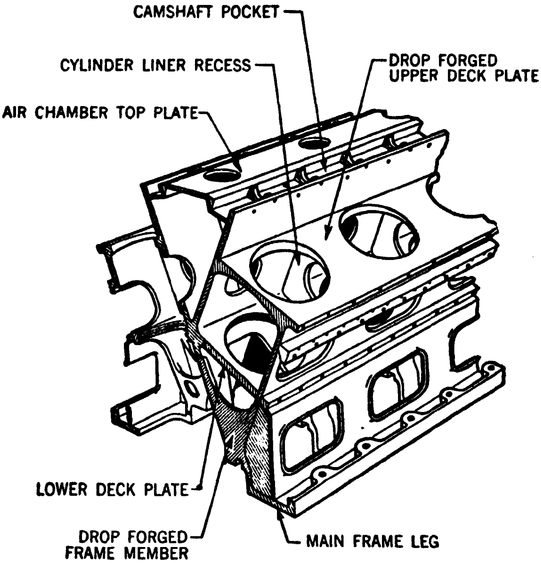


FIG 4-2. Section of welded engine frame. Many high-speed engines now have frames of welded steel plates located where the loads are put on the engine. This is a Winton engine frame. Many marine engines are designed with welded construction

and the thin sections hard and unmachinable. When the silicon is increased to keep the light sections soft, the heavy sections are likely to be open-grained and spongy. Also, straight silicon irons *oxidize* readily at elevated temperatures and the combined carbon tends to break down into graphite and ferrite. Such a reaction involves an increase in size, or *growth*, and a decrease in strength. It is advisable therefore that the silicon be kept as low as possible.

The addition of nickel, while supplementing the silicon in reducing chill, refines the iron and produces greater structural uniformity. It also reduces the tendency for draws and slightly

increases the combined carbon, making the iron harder and more wear-resistant. However, such irons are readily machinable, and a fine surface finish may be produced. Through the use of nickel, the casting of many difficult sections is made readily possible that would otherwise be extremely difficult, if not impossible.

Chromium and nickel, in suitable proportions, develop excellent characteristics in iron, particularly where resistance to wear and growth are important factors, such as heads, pistons, and cylinders.

Molybdenum has a beneficial effect on cast iron. Its effect is intermediate between nickel and chromium in that it tends to develop stability, without undue chilling tendencies, and a compact graphite formation which is very beneficial in resisting fatigue. It may be used alone or in combination with nickel.

Strains may be set up in irregular-shaped castings as a result of differences in rates of cooling throughout the mass. If the castings are left in the sand, until fairly cold, the *natural annealing* effect of the slow cooling should effectively relieve the stress and strains. If not relieved prior to machining, the piece may be distorted in service. A safe procedure in cases where distortion may affect the operation of the unit is to anneal definitely the casting in a suitable annealing furnace, a practice followed by most manufacturers.

With facilities for annealing available, the additional step to heat-treating is comparatively simple. The desire for greater hardness in parts subject to wear, such as cylinder liners, has resulted in a development of irons susceptible to heat treatment. Such irons ordinarily contain from 2 to 5 per cent nickel, plus the proportional amounts of chromium and, like steel, may be heat-treated to Brinell figures of 400 or better. Although comparatively strong and tough, the materials are still cast iron and as such are inherently brittle. The treatment is best confined to pieces of regular sections because of the possibility of cracks developing through the strains set up during *quenching*. The machining procedure follows that of treated alloy steels. The softer pieces may be machined after treatment; whereas the harder ones must be first roughed out, treated, and then finished off by grinding. Iron, heat-treated for hardness, softens under heat, and hence will lose its hardness if operated at a temperature in excess of that used in tempering to develop a given hardness. However, the range is not seriously restricted

as temperatures around 900° F are usually employed during heat treatment.

The necessity for greater resistance to corrosion and high temperatures and greater stability has led to the development of a still more highly alloyed iron known as *Ni-Resist.* This iron contains about 15 per cent nickel and 7 per cent copper, with chromium varying from 1 to 6 per cent. The matrix is *ausenitic* in type, somewhat similar to that of the 18 and 8 steel of the stainless type. It is nonmagnetic, tough, nonhardenable by heat treatment; and if the chromium content is not too high, it is reasonably machinable. Its coefficient of expansion is relatively high, approximately that of aluminum. Although comparatively soft, it will work harder than other metal in service and develop appreciable resistance to wear. It can be produced in the foundry under standard conditions, although irregular sections may be difficult to cast because of shrinkage.

Steel castings. Steel castings, because of their combination of high strength and ductility, when suitably treated, and because of the fact that they can be produced in shapes not permissible in forgings, serve many useful purposes. Like any other material, they have their peculiar faults, which must be guarded against if trouble is to be avoided. Blowholes, shrinks, and segregation are the principal weaknesses, and unfortunately they are sometimes difficult to detect.

Cast steel lacks a constituent like graphite to counteract the effect of metal shrinkage, and unless uniform sections are provided or provisions made to feed the heavier sections while cooling, shrinkage cracks may result. As a result, complicated castings with irregular sections are much more difficult to obtain in steel than in iron. The beneficial effect of heat treatment on a carbon steel casting is illustrated by the following table:

TABLE 4-1

EFFECT OF HEAT TREATMENT ON CARBON STEEL CASTING

Conditions	Tensile lb in.^{2}	Yield lb in.^{2}	Elongation %	Reduction %	Brinell	Impact
As cast	74,800	37,500	19.5	29.0	156	17
Normalized	75,650	42,000	25.5	44.0	143	21
Treated	84,200	57,400	31.5	65.0	160	44

Alloy-steel castings are now being produced in almost as many varieties as obtainable in bar stock, and if suitably heat-treated serve a useful purpose. The greater tensile strength permits lighter weight, and the lighter sections tend to promote soundness. Stainless steel castings of many grades are also being produced, although their use in Diesel engines, because of cost, is rather limited at the present time. These steels are difficult to cast, and because of their high solidification shrinkage, usually require special pattern equipment and simpler designs.

Structural steel. Aside from straight carbon steels, the number of alloy steels available for structural purposes is legion. Many have been standardized by the SAE and are designated by their numbers, while others, the special products of some steel company, are sold under brand names. Although many of these steels have distinctive characteristics and serve useful purposes, nevertheless the number could be materially reduced with profit to the industry. As a matter of fact most of the trouble encountered, where steel is supposed to be at fault, is due to improper heat treatment rather than any inherent fault in the type. The fact that better results can usually be obtained in the long run through the suitable conditioning of a standard type of material, rather than through the hit-and-miss tries of special types, cannot be too strongly emphasized. All steels, particularly the alloy types, must be suitably heat-treated to develop their maximum physical characteristics.

Special steels. Numerous types of special alloy steels are also available. Such steels are of value for their ability to perform some special purpose well, such as air hardening without distortion, resisting wear, or standing up under corrosion. *Chromium* is the principal element in most of these steels, although nickel, molbydenum, vanadium, copper, silicon, and aluminum may play important parts.

Stainless steel or iron, a low-carbon 12 per cent chromium material, has many useful purposes. If properly treated, it has excellent physical characteristics and reasonably good resistance to corrosion. With properties comparable to those of SAE 3140 steel, it may be safely used where strength is required. For greater stability, under severe conditions, one is forced into the austentic steels carrying appreciable percentages of chromium and nickel with an additional element such as silicon. Many modifications are available.

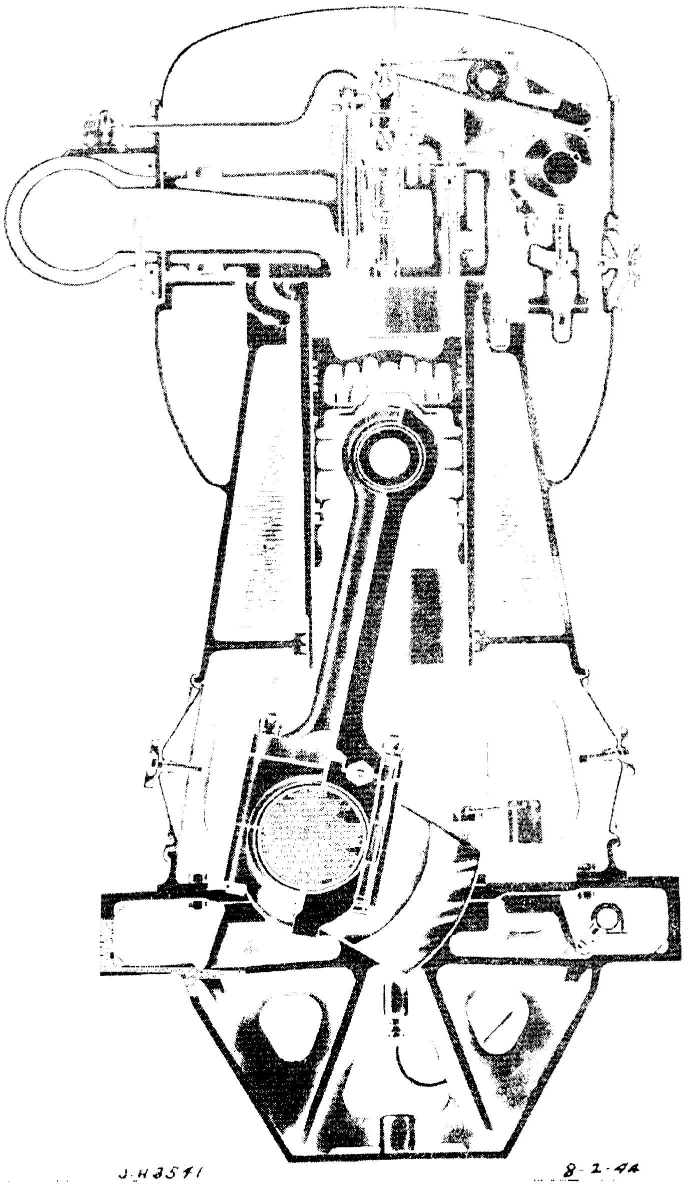


FIG. 4-3. This heat-treated cylinder and frame (Hendy Series 50) Diesel cylinder block is fabricated and constructed of welded heavy steel plate, and stresses relieved with heavy cross members to support the main bearings. The cylinder block is bolted onto the base and carries the cylinders and all other parts of the engine.

Nitralloy, a special type of steel that may be processed to develop remarkable surface hardness, stands up well under severe conditions of wear. Two grades, the chromium-aluminum, having exceedingly high hardness, and the chrome-vanadium, with less hardness, but greater toughness, are in general use. The material retains its hardness through the temperature range encountered in Diesel engines, with the possible exception of the exhaust valve. It is very resistant to both atmospheric and *alkaline* corrosion, but not to *acids*. However, from records available it would appear that neither type is seriously attacked by the usual fuels.

Forgings. Forgings assure a sound section in the finished piece, free from blowholes and draws. Further, the quality of the stock used may be carefully checked before the forgings are made. However, drop forgings have their peculiar faults in the form of *laps* and *cold shut*. Fortunately these, unlike blowholes and shrinks in castings, can usually be detected readily by visual inspection. Drop forgings have an advantage over castings in that they may be made in light sections and still retain a clean-cut appearance.

Unless care is exercised in forging, the structural condition of the material, despite the beneficial effect of the added working, may be utterly ruined. If steel, for instance, is soaked at a high temperature *in an oxidizing atmosphere, the gases penetrate* the section and *oxidize the grain boundaries*. As a result the material becomes extremely brittle; and the condition is not corrected by any normal heat treatment. Some indication of this condition, if present, can be detected by examining the pickled surface, but the only safe method is by an actual fracture test.

As the material is elongated under the hammer, it tends to accentuate the directional grain flow normally found in bar stock. The ductility, when tested with or across the direction of flow, is usually much lower than when tested with the grain. In some pieces, such as a crankshaft, the directional grain is an asset, providing the fibers are continuous throughout the length, while in other parts, such as gears, where the direction of the stress application is continually changing, the directional grain is not. The directional tendency may be broken up by suitable cross forging, and the desirability of so doing must be considered when designing stressed members.

Forgings, like castings, must be properly heat-treated to

secure uniformity of structure, relieve strains, and to obtain the maximum conditions of strength and ductility. Light sections should be fully heat-treated while heavy sections may be normalized and annealed.

Aluminum alloys. Aluminum in its pure condition, like iron, is a soft, tough metal, but when suitably alloyed with copper, magnesium, or other elements develops many desirable physical characteristics. All of the alloys may be cast into use-

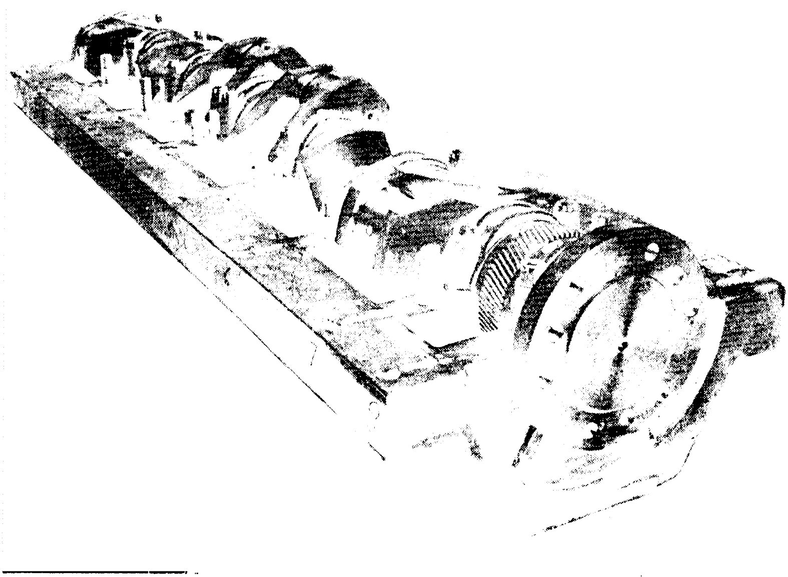


FIG. 4-4. This crankshaft and engine bed (Hendy Series 50) has a welded steel base designed for rigidity and heavy cross members to support the main bearings and keep the shaft in perfect alignment.

ful form, while some, in addition, may be forged, rolled, or extruded. Many of the alloys are naturally soft and tough, others are comparatively strong but brittle, whereas some, when heat-treated, develop a good combination of strength and ductility. In the wrought form, treated, tensile strength of over 50,000 psi with elongations of around 20 per cent may be obtained. The modulus of elasticity, however, is only one third that of steel.

The aluminum alloys are unique in that their physical characteristics tend to run to extremes, and unfortunately, some of them conflict in their usefulness. Light weight, the

outstanding characteristic, is offset to a certain extent by low modulus. The high heat conductivity in many cases is an advantage, whereas in others, it is not. The high coefficient of expansion, common to most of the alloys, does not work out well with cast iron. The bearing characteristics of aluminum are not good, and the resistance to *alkaline corrosion* is low.

Although aluminum alloys are regularly used for many purposes, particularly where weight is a consideration, the selection of the most suitable type must be given careful consideration in the Diesel engine.

Bronzes. The various types of bronze alloys available under various standard specifications are exceedingly numerous. In general, all the bronzes are resistant to atmospheric corrosion, and with suitable modification can be made to resist attack of any of the conditions encountered in the Diesel engine work.

The majority of the bronzes are used in the cast form, although many may be die-cast and forged. The ease with which they can be fabricated and their resistance to corrosion tend to make them attractive to the engineer as structural members, but these advantages are in most cases offset by the low modulus of elasticity, weight, and comparatively high cost. In bearing service, however, their use is well established.

The bronzes, as a class, *oxidize* readily when molten, and unless care is used in the foundry to prevent such a condition developing, their physical characteristics may be seriously impaired. Such a condition will show up in a tensile test, largely through low ductility, although the condition can be readily detected by means of a simple fracture test. Oxidation in a tin-bearing bronze is particularly bad, as any of the extremely *hard tin oxide* formed and scuffed out in running may seriously score the shaft. The temperature of casting also has an important effect on the strength and ductility of many of the bronzes.

Monel metal, because of its combination of strength when suitably processed, its higher modulus of elasticity, its wear resistance, and its ability to withstand most forms of corrosion, is an excellent intermediary material between the bronzes and steel. It is readily available in wrought form, but sound castings, if irregular in shape, are not so easily obtained.

Bearing metals. An ideal bearing metal consists of two constituents. There should be a tough base metal that acts as a body or matrix to carry a hard, finely dispersed element or

alloy, which, in turn, in the aggregate acts as a bearing surface. The hard constituent resists wear; the soft constituent provides the degree of conformity necessary to resist any local high spots, and through its natural tendency to recess assists in maintaining the oil film. Cast structures, because of the natural arrangement of the grains, are usually better than those that have been mechanically worked. Bearing materials should have a low *coefficient of friction,* although under running conditions the actual friction is dependent on the pressure, speed, temperature, and lubrication.

The white metals, which are essentially alloys of tin, lead, copper, and antimony, are the commonly recognized bearing metals. The tin-base or babbitt metals have a better combination of strength and *plasticity* than the lead-base alloys, and hence are more generally used in Diesel work, even though more expensive. However, considerable work is being done with the lead-base alloys, and some worth-while results have been obtained.

The white metals are sensitive to variations in compositions, and small changes in the hardening elements or some impurity may either render them brittle or lower their melting points appreciably. The metals oxidize readily when molten, and hence extreme care must be exercised in both the melting and pouring operations.

The high-lead bronze, known as the high bronze, with or without small percentages of tin, is closely associated in application with the white metals. Such alloys do not conform to the true concept of a bearing metal, but actually the reverse, in that the copper matrix is harder than the dispersed load. However, they seem to function well within their working range, which is the fusion point of the lead. A rather high coefficient of expansion and possible *lead segregation* are the two features most likely to give trouble in the use of this material.

The harder cast bronzes, carrying an appreciable amount of tin, lead, and possibly some *phosphorus*, are good bearing metals when operating against hardened steel. Such bronzes have good strength and resistance to wear, but good alignment and adequate lubrication are essential to their satisfactory performance. If the oil film falls, the resulting seizure is likely to be disastrous to the shaft. Numerous types of bearing bronzes are available, and the selection is largely a matter of past experience.

The structural condition of the material used should be checked by fracture test.

The beryllium-copper alloys, capable of being treated to a Brinell hardness of around 400, are being advocated as bearing metals. Their use, however, has not spread sufficiently to warrant satisfactory conclusions being drawn as to their probable performance.

Chromium plating. "Hard" chromium plating is being used rather extensively because of its excellent *wear-resisting* characteristics. This type must not be confused with the "decorative" plating, wherein an extremely thin film of chromium is applied, more or less as a lacquer, to preserve the copper-nickel plating underneath.

This hard plate is applied directly to the base material in thickness varying from one half to three thousandths of an inch, depending on the conditions of the service. Either steel or bronze, as a base material, plate very satisfactorily. Cast iron, because of the graphite, is not such a suitable base.

When properly finished by grinding and polishing, the plated chromium presents a hard, low-friction, wear-resisting surface suitable for bearing purposes. The plate, although sufficiently tough and adherent for normal bearing conditions, is not suitable for use under shock or extreme pressure such as might be applied through high spots in a hard-packing material. Chromium works best as a bearing material when mated with a soft packing such as a leaded bronze. Cast iron, particularly, if it contains free *cementite*, is likely to score the plate through an indentation of the softer base material.

The hardness of plated chromium is dependent on the presence of the *hydrogen deposited during plating*, and if heated sufficiently high, in the range of 1400° F, the plate becomes soft.

Unless the conditions of plating are carefully controlled, strains are set up in the deposited metal sufficient at times to cause it to crack. Such cracks may, in extreme conditions, extend through the entire depth of the plate, in which case the tendency to spall is increased and the resistance to corrosion decreased. However, because of a better knowledge of the variables responsible for this tendency, the plate being produced now is comparatively free from these defects and should perform well in service.

Chromium works well on shafts, pins, and similar parts where good bearings and adequate lubrication are provided.

It also functions well as a protection against *galling* on liners or plugs that are assembled with a close fit and that may later have to be removed. Worn surfaces may in some instances be built up with chromium and satisfactorily *salvaged*. Further,

Fig. 4-5. Welded steel block of Hendy Series 50 engine.

chromium-plated surfaces, when worn, may be renewed at small expense.

Most of the problems encountered in operation are fundamentally ones of design, but for economical reasons they often become metallurgical. There are no complaints on this score, but it is believed that a more careful study of the details, with a view of eliminating localized overstress and providing better lubrication, will eliminate most of the troubles now encountered. As a matter of fact, where such considerations are given, the performance of the Diesel engine is most reliable, and many

engines are building records that rival those of the older reciprocating steam engines of early days in length of engine life.

Welding. Welding has been extensively used in the construction and repair of Diesel engine parts. The technique of welding has reached the point where it has ceased to be a metallurgical problem so far as steel construction is concerned. With suitably *coated rods* of the same composition as the base material, a skilled operator can lay welds having substantially the same physical characteristics as the original plates. A low-temperature, strain-relieving treatment, however, should follow the welding of all complicated sections that are likely to be subjected to *vibratory stresses.* For the higher-carbon and alloy steels, the temperature should be carried above the critical transformation point of the material, so that, in addition to relieving the strains, there may be a refinement of the coarse grain developed during the welding operation.

Cast iron can be welded under favorable conditions, but on the whole the results are not overly satisfactory. The structure of the iron matrix is badly upset, and *incipient cracks* may develop in and about the weld unless the operation is most carefully controlled.

Design, in built-up structures, is all important and far outshadows the metallurgical considerations. The liberal fillets in castings, to avoid shrinkage draws that tend to develop at sharp corners or sudden changes in section, and reasonable thickness of metals, to prevent misruns, automatically serve to stiffen the structure and smooth out any tendency for concentration of stress. As a result, castings are usually very rigid because of foundry considerations, and therefore many of the engineering problems disappear. In built-up structures, however, the reverse is true. The material is strong and tough, and as far as the static stresses are concerned, it can be used in light sections, and the fillets are naturally small. Unless proper precautions are taken, therefore, to stiffen such a structure by suitable bracing, flexing is likely to occur in service and with it a concentration of stress in the welds. Such a condition usually results in ultimate failure not because of faulty weld, but because of improper design. Where weight is an important consideration, welding undoubtedly provides a means of keeping it down.

While welding of structural members represents sound practice, the building up of worn or *undersized working parts*

is open to question. Many failures have been traced to this cause. Active parts are usually made of medium-carbon or alloy steels, which do not weld as satisfactorily as the low-carbon structural steels. The higher strength materials are more likely to be *spongy* when welded, and a suitable structural condition after welding can be developed only by heat treatment. Such a heat treatment, accompanied by surface scaling and distortion, is of course ruinous to a finished piece.

Where the stresses are inherently low and welding is not being performed at a change in section, the procedure may not be harmful if done under competent supervision, but otherwise it should be looked upon with disfavor.

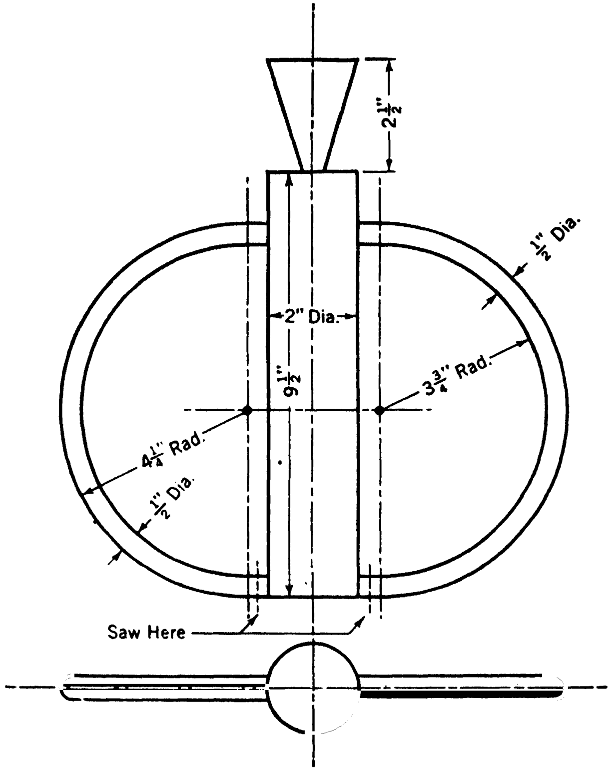
Failures in Diesel Engine Parts

Failures due to material and design. Failures may be due either to faulty design or faulty material, and sometimes to both. A material may be considered at fault if some irregularity, not common to a properly conditioned piece, has contributed to failure. The irregularity may be a cold shut, segregation, poor structure, deep tool mark, or a similar defect. The design or selection of material may be considered at fault when a sound homogeneous piece, properly conditioned, fails for no apparent reason. As a rule, occasional failures denote faulty material, whereas repeated failures such as repeated cracking of cylinder heads or breaking of crankshafts in service tend to throw suspicion on the design.

Although it is not practical to establish positively the soundness of each piece of material, nevertheless, with close supervision over the methods of manufacture and adequate inspection in the shop, the chances of faulty material getting into service are not great.

Changes in material can be made without any outward evidence of its having been done. Changes in design, however, usually involve not only the piece actually defective, but other mating parts as well, to say nothing of tools, fixtures; the interchangeability of parts must be maintained. Consequently, where a trouble may be corrected through the substitution of another type of material, it is logical that such a change should be made. The result, however, is that many problems that are fundamentally problems of design automatically become problems of metallurgy. Although a piece of alloy steel may be sub-

stituted for carbon steel or a piece of Nitroalloy for hardened tool steel, the extent to which changes of material may be made is comparatively limited, and considerable thought must be given to the original design and selection if trouble is to be avoided. Operating problems in which the material is involved



Courtesy International Nickel Company.

FIG. 4-6. A test casting. After heat treatment, the circular side bands were severed at the top next to the central body. Any residual casting stress would cause these side bands to expand or spread apart from the central body. This spread is considered a measure of the residual stress and is plotted in Fig. 4-7 as expansion.

may be grouped under three general subjects: breakage, excessive wear, and corrosion and oxidation.

Breakage. Generally speaking, breakage, assuming that the material is correct, is of course the result of *overstress*. It has been found that pieces made of excellent material and operating under low-calculated values of stress occasionally fail. The cause, although not always apparent, can usually be traced to

the *concentration of stress.* In irregularly shaped pieces the average stress is of little practical importance, as failures are the result of maximum stresses. Each type and condition of material has a certain maximum stress value, which is called the *endurance* or *fatigue* limit, under which the piece will withstand a definite number of applications of stress without failure. If this stress is exceeded, however, even though over a minute area, a crack will ultimately form and failure occur.

Stress concentrations may be due to sudden changes in section, deep tool marks, inclusions, or anything that tends to break the smooth continuity of the section; or they may be the result of *misalignment,* and excessive vibration.

While the *intensity of the stress* concentrations resulting from changes in section may be difficult to calculate, their probable location and approximate magnitude can often be determined through the use of bakelite models and a polariscope. Under the action of polarized light, the stress distortion in section and distribution in a section of bakelite may be observed by color variations in the material. Such a method of investigation is very instructive and simple.

It is evident that stress concentrations are the most prolific source of breakage, and effort should be made to keep them to a minimum.

Fillets at shoulders in stress sections are recognized as essential, but slow tapers, if practical, are better in design practice. Similarly, splines, distributed around the surface of the shaft, are better than a single key. Whitworth threads are better than V threads because, in the event of bending, the stress concentrations at the roots are much lower with the smooth radius than with the sharp notch. For the same reason, studs with the body turned down to the root diameter of the thread are much better than those with full-sized bodies; the reduced diameter of the body, aside from reducing the stress at the roots of the thread, provides greater flexibility along the entire length and less tendency for concentration at any point.

There is a tendency in design to lay too much emphasis on the elastic limit and tensile strength and not enough on the so-called toughness of the metal. The fatigue or endurance limit, obtained with a smooth symmetrical specimen running *without impact,* seems to be a function of the ultimate strength. In actual service, however, such conditions really rarely exist, and aside from the irregularities of operation, parts are seldom

regular in section. Stress concentrations, not considered in the fatigue limit, are therefore an important consideration in actual practice. If the material is brittle, a crack once started will progress very rapidly, whereas if the material is tough, the condition of overstress may be relieved through a local readjustment of the grains in the area of the stress. Even though the formation of a crack cannot be avoided, the tougher material may seriously retard its progress and delay the eventual failure of the part.

Unfortunately, the toughness that tends to retard the formation and subsequent development of a fatigue crack is not indicated by any available mechanical test. Pieces having the same tensile and impact values may show decided differences and performances in service. The microscope is the most useful method of checking certain characteristics. In steels, for example, a coarse-grained, pearlitic structure, under many conditions, is not so resistant to fatigue as a normalized structure consisting of a fine-grained intimate mixture of ferrite and pearlite. The soft, tough ferrite seems to provide the mobility that is so important in resisting the spread of cracks. High-tensile alloy steels, not sufficiently treated, may fail more quickly than a comparatively soft, low-strength material. Good balance must be maintained between strength and toughness and when some of the more highly stressed parts are specified the microstructure is usually specified.

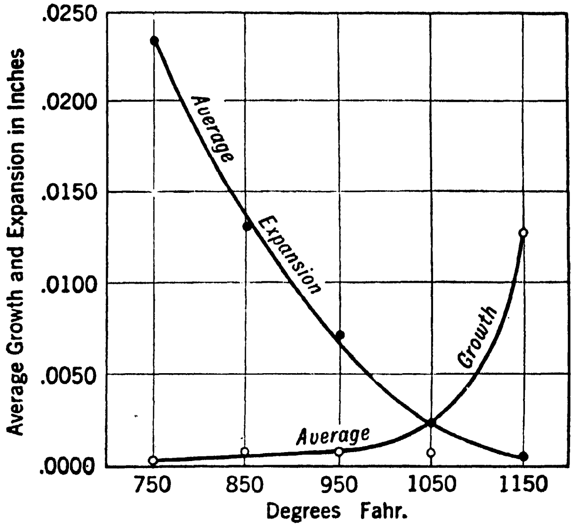
Excessive wear of engine parts. The important class of failure, known as excessive wear, may be due to faulty material, although high pressures, due to abnormal conditions of operation, and inadequate lubrication are more likely to be the real causes.

Wear is the result of rubbing, metal-to-metal contact of two surfaces. If the structure and surfaces are continually separated by a thin film of oil, wear is largely eliminated and is reduced to a mere matter of erosion and possible chemical changes. When the film is periodically broken, however wear is usually very rapid.

The normal working pressures are readily calculated and may be amply provided for in the original design of the engine. Abnormal stresses due to *periodic vibrations* or the whipping of shafts are not so easily determined, and the resulting pressures may be sufficiently high to break the oil film. Misalignment, surface finish, and dirt are potential sources of wear, although

these may be really due to neglect and can be controlled, and hence they should not be serious factors in the actual operation of engines.

Numerous types of metals suitable for operating in contact with one another are available, and if worked within their corresponding limitations, a satisfactory performance is assured. The selection of the most suitable combination of metals should



Courtesy International Nickel Company.

Fig. 4-7. Effect of annealing one to three hours on growth and expansion of stress-test bars. The less the expansion the greater the relief of internal stress (Walls and Hartwell). Tests conducted on cylinder iron containing about 3.30 per cent total carbon and 2.30 per cent silicon. While a temperature of 1160° F will relieve practically all stress, appreciable structural alterations occur. The best temperature to relieve maximum stress for this silicon content with minimum alteration of structure is about 900° F. The effect of heating above 900° F is further evidenced by the softening shown in Table 4-1.

be given careful consideration, since in addition to obtaining long life as a whole, the conditions should be such that the least expensive of the two pieces in contact will wear most. A hard-bronze bushing may readily score and gall a soft-steel shaft, while a soft bushing will not. A soft material in the presence of dirt, however, may load up and, as a result, lap away a hard material very rapidly.

In order to maintain a good oil film, if proper alignment, finish, and rigidity are assumed, it is necessary that the elements

of temperature, pressure, feed, clearance, and oil viscosity be held within limits. The viscosity of an oil decreases with an increase in temperature, and if thinned down too much, the resulting film may not be sufficient to carry the load. The pressure, taking into consideration the surface velocity, should be kept within the limits which experience has proved most satisfactory for the metal combination employed. The clearances must be adjusted to suit the viscosity of the oil being used. If the clearances are too small, the film will not be established; and if too large, the oil may work out and not maintain a continuous film. The arrangement of the feed must be such that the distribution, developed in the unloaded area, is uniformly directed to the zone of pressure.

Perfect lubrication is never attained in actual operation; hence, metal to metal contact does occur, even though only at the time of starting, and consequently, wear is inevitable. Through careful utilization of available materials under proper conditions of operation, good lubrication may be maintained at a very low rate and comparatively long life of the parts realized.

Corrosion and oxidation of parts. Corrosion and oxidation are more likely to be metallurgical problems than wear or breakage, for in spite of all the designer can do, the materials must work in contact with the corroding and oxidizing medium.

The exhaust valve is probably the hot spot in all internal combustion engines and usually reaches in operation a rather high temperature. The other parts, however, which can be more readily cooled, do not become excessively hot.

The Diesel engine, however, unlike the gas and gasoline engine, draws in air undiluted with fuel and compresses it to a temperature at which the oxygen becomes active before the fuel is admitted. The oxygen, therefore, in the absence of fuel, is free to attack the iron or other materials forming the surface of the combustion chamber. Fuel oils for Diesel engines are usually not so highly refined as gasoline, for example, and usually contain appreciable amounts of sulfur which become rather active chemically.

Even though the injected fuel is carefully atomized, there is in many cases a tendency toward *flame impingement.* In the case of iron pistons, if the heat is localized, particularly at the center, growth takes place due to a combination of *temperature variations*, and a crack may develop. The upper portions of the

liners, also, wear more rapidly than the lower portions, largely because of this same condition.

Design application to engine parts. Having shown the relation of design and metallurgical considerations to Diesel engine parts, and the influences that cause breakage, excessive wear, and corrosion, it is in order to consider the material application to the various engine elements that go to make up the assembled structure.

Engine frame works. Important considerations in the design of a frame are strength, rigidity, and weight. Frames may be made of cast iron, aluminum, or welded steel. The ease with which cast iron can be fabricated and its comparatively low cost account for the majority of the engine frames being made in that material. Welded-steel frames, however, are being given consideration because of their possible lighter weight.

If made of cast iron, the composition may vary all the way from a straight silicon to a high-test alloy, depending on the particular requirements of the service. Care must be exercised to keep the sections reasonably uniform to prevent *casting shrinks* and strains and to avoid ultimate breakage.

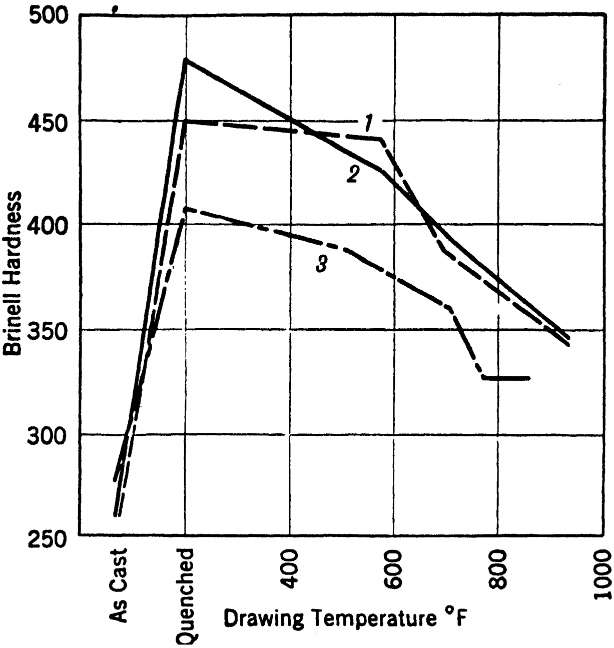
Welded frames are usually made of soft steel and finally annealed. If made of high-carbon or alloy steel, the entire frame is heat-treated to eliminate the brittle area which develops near the welds and to develop fully the advantage of the higher-strength materials. Rigidity must be obtained if weld failures are to avoided. The frames of small units, where the minimum weight is desired, may be made of aluminum.

Cylinders. In the smaller units, the cylinders may be integral with the frames while in the large units they are made separate. When the cylinder is to provide its own liner, considerable attention is paid to the type of material used to obtain the minimum wall wear, otherwise the casting may be made of any type of material that will develop the required strength. Care is usually taken to see that a smooth water-jacket surface, having as little attraction for scale as possible, is provided. These castings are usually given a low-temperature anneal to relieve the strains and to prevent subsequent distortion, or they may be seasoned by aging.

Liners. Cylinder wear is an important consideration in the operation of a Diesel engine. The rate of wear is dependent on one or more of several factors such as lubrication, ring pressure

from the gas reaction, dirt, oxidation, and type of material used in the liner.

Lubrication and ring performance are matters of design, while dirt, if a factor, may be controlled to a large extent by the use of a suitable air filter. Oxidation, while influenced to a



Courtesy International Nickel Company.

FIG. 4-8. Effect of quench and draw back on Brinell hardness of nickel-chromium cast-iron piston rings (Hurst).

No.	*1*	*2*	*3*
Total Carbon	3.36	3.35	3.39
Silicon	2.30	2.07	2.09
Nickel	1.32	2.56	3.37
Chromium	0.44	0.95	0.61
Quenching medium	Oil	Oil	Air

This graph illustrates results for an air-quenched composition, which is an alternate for castings too delicate for oil quenching, such as piston rings individually cast.

considerable extent by the fuel used, is more dependent on *temperature attained*, and hence varies with the conditions of cooling. Thin liners, for this reason, are better than thick ones.

Liners may be either of the dry or the wet type. The dry type is mostly confined to the small sizes because of the thicker

section of the two walls and the difficulty of providing the close fit essential to good heat conductivity. The metal conditions are much more difficult to control in the dry type than the wet type because of the larger castings involved.

To resist wear the liner material usually has a reasonably hard, dense structure, with good bearing characteristics and stability under the conditions of operation. To meet these requirements the majority of liners are made of cast iron, although some are being made of steel and hardened Nitralloy. In cast iron, this necessitates low silicon to resist growth, low total carbon to develop denseness and eutectoid ratio in combined carbon, and low grain size to develop resistance to wear. Nickel and, if the size will permit, chromium are helpful in obtaining the desired condition.

Because of their symmetrical section, liners may be cast centrifugally and a denseness and uniformity of structure obtained not easily reproduced in a sand casting. If cast in the ordinary manner, the mold is usually placed vertically, and the casting is machined so that the bottom or drag end, as cast, will be uppermost in the cylinder. In this method, the denser metal will be placed at the point of maximum wear and excellent results are obtained.

Under usual conditions, the Brinell hardness of a straight silicon iron cannot much exceed 180 and that of a regular chrome-nickel, 220, without free carbide being present. Such carbides are objectionable because of their abrasive characteristics when scuffed free of the iron. If a higher hardness is desired, it is obtained through the use of a heat-treated iron. Nickel, plus some chromium, in a cast iron, when suitably treated, will permit the development of an excellent structure with a Brinell hardness figure of 400 or more.

A 60-point carbon steel, normalized or heat-treated for greater hardness, has, when adequately lubricated, proved satisfactory as a liner. This section can be used and excellent cooling thereby obtained. Ni-Resist cast iron offers excellent possibilities as a liner material in combination with aluminum, as the coefficient of expansion is about the same in both materials. It is rather resistant to attack from either heat or normal gases, and its soft, *austenitic matrix* tends to take on a work-hardened glaze that makes it resistant to wear.

Nitralloy steel, nitrided, is being used on many of the smaller engines with reported good results, and aside from the

possible attack where high sulfur fuels are used, there is no good reason why its use should not come to be established.

Cylinder heads. Cylinder heads are usually made of cast iron, while steel may be used on the larger units and aluminum on the smaller units. The head must be sufficiently strong to resist the combustion pressures and have the necessary stability to resist the attack of the air and gases in operation.

Head sections of the single-acting engines are usually rather complicated because of the manifold passages and water-cooling chambers. As a result, the heat stresses may be rather

FIG. 4-9. Crankshaft (Hendy Series 20) cast from an alloy steel with hollow pins and journals. Lubrication to the main journals and crankpins is through inserted tubes. All bearings are removable through side inspection doors.

high, and this fact must be taken into consideration in drawing up the design and selecting the material. When made in cast iron, the heat may provide its own valve seat, and if made in aluminum, some other material is inserted. Aluminum bronze and Ni-Resist are frequently used for a valve seat.

The water jacket is usually smooth and clean in order to resist the adherence of scale. Attention is paid to the cooling water used with aluminum, otherwise porosity, the result of corrosion, usually develops.

Pistons. The conditions imposed upon the piston, like those of the liner, are very severe. A piston should have light weight, strength, good bearing characteristics, an expansion rate comparable to the liner, good heat conductivity, and resistance to the attack of the hot air and gases. No single material will meet all these requirements; hence the one selected must be a compromise. Cast iron is largely used, however, because its coefficient of expansion may be the same as that of the liner; its bearing characteristics are good; and it is comparatively

inexpensive. The pistons, like the liners, should be made of a dense, fine-grained material. The majority are sand cast, although some of the smaller sizes may be made in semipermanent molds.

The straight silicon irons oxidize readily in contact with the hot oxidizing gases, and if there is a tendency for a local hot spot in the center of the piston, it may fail rapidly through *growth* and fatigue. The use of nickel or nickel-chromium iron retards this type of failure. In the event of seizure with a cast-iron piston, the resulting damage to the liner may be rather disastrous; and only care in operation will prevent accidents of this sort.

Aluminum, because of its light weight, is being used extensively for pistons in the smaller high-speed engines. It may replace cast iron on larger engines where water cooling would otherwise be necessary although the experience in this application has not altogether established complete operating data. The rather high coefficient of expansion is difficult to control in larger units. It may be fairly well accomplished in the smaller engines through the use of Invar rings, and therefore the tendency to slap when cold may be somewhat eliminated. In some cases, the expansion is taken care of through the use of a split skirt.

Aluminum has the disadvantage of being a comparatively poor bearing material, and hence requires good lubrication in a place where good lubrication is not easily maintained. Also, being a soft material, it loads up with dirt and acts as a lap on the liner. Seizure is not so severe, however, as with an iron piston.

Pistons of the aluminum type are usually made of a copper-magnesium-nickel alloy, heat-treated to develop a Brinell hardness of around 120. Because of the low modulus, there is a tendency for distortion under load, which must be taken care of with additional clearance in the vicinity of the piston pin.

Steel is not generally used as a piston material because of the difficulty in obtaining a satisfactory casting and its *tendency to gall* under failure of lubrication. Piston heads may be made of steel and used with iron skirts, however.

In a later chapter, what appears to be a solution of the piston design problem will be discussed in some detail.

Piston rods. One of the most likely sources of trouble in the double-acting engine is the piston rod. The diameter of

the rod must be kept small in order to interfere as little as possible with the capacity of the lower combustion chamber. This means that the working stresses must be high. The rod must also act as a carrier for the piston-cooling medium. If the rod is hollow bored to permit a circulation of the cooling liquid, the temperature gradient stresses may be quite appreciable. If the rod diameter must be kept smaller than would otherwise be necessary, the corrosion and the resulting stresses are high. If the circulating medium is water, the danger of corrosion is great. Corrosion not only lowers the endurance limit of the steel, but it causes pits that localize the stresses, further accelerating failure.

On account of the high working stresses and the packing wear, the natural tendency is to make the rods of alloy steel, suitably heat-treated to develop strength and hardness. However, such rods do not seem to stand up as well as those made of softer steels, because they apparently lack that inherent toughness that is so essential to resisting the progressive type of failure. Either straight carbon or some soft alloy of the nickel-manganese type seems to be used most often in practice and works out best.

Piston rings. Piston rings are usually made of straight silicon cast iron regardless of the type of liner used. Excellent results have been obtained through the use of nickel iron, and its use for this purpose should increase. Steel, of approximately spring temper, may be used in smaller sizes. Experiments have been conducted with other types of iron and hardened steels, but the results have not been sufficiently outstanding to bring about any general change from cast iron.

Piston pins and bushings. The chief requirements of the piston pin are that it be light, have sufficient strength to carry the load, and have the proper hardness to resist wear. With adequate lubrication and a suitable fit, a piston pin should resist wear indefinitely. Pins may be made of hardened tool steel. However, they are more usually made of carburized and hardened steel. Nitrided and chromium-plated pins have also been used with some success. A smooth, highly finished surface is essential to long life.

Bearing bronzes of the composition type are most generally used for bushings, although copper-lead seems to be meeting with some favor because of the smaller damage to the pin in the event of seizure, and this type under normal conditions works satisfactorily.

Crankshafts. One of the most desirable conditions or characteristics in a crankshaft is that the *directional grain flow* of the

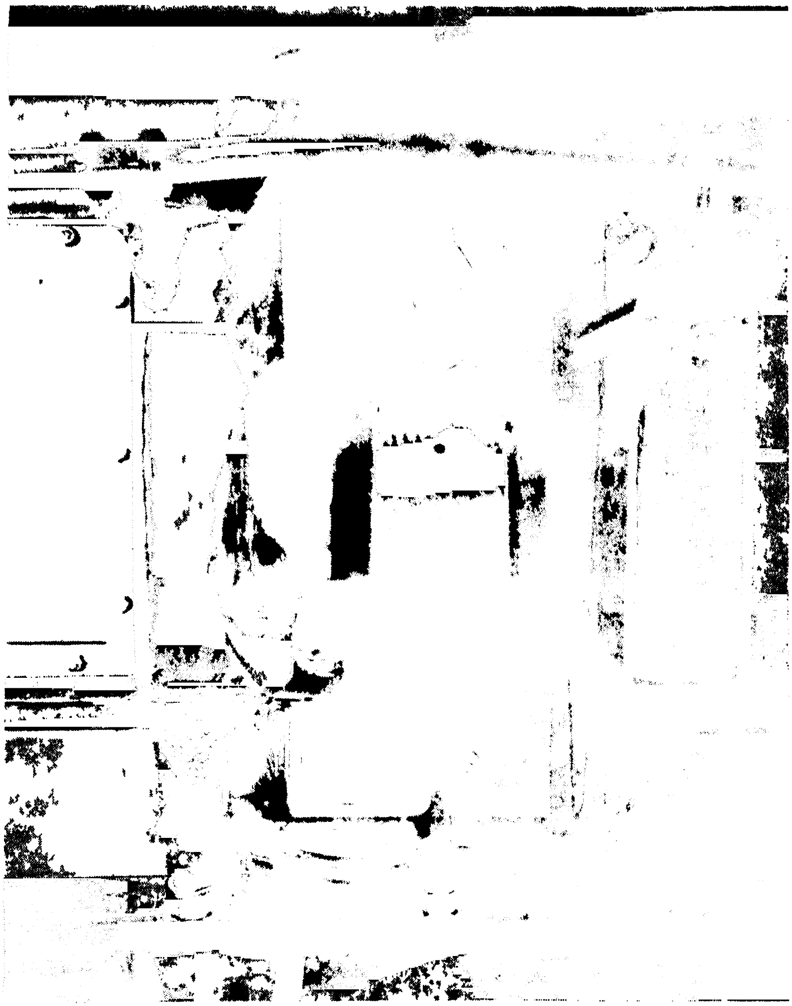


FIG. 4-10. Wreckage of a 1000-hp Diesel engine where the piston and rod were thrown through the side of the engine while under load at full speed. Deep scores and cuts can be seen in the crankpin. It was believed that the connecting rod bolts were overstressed and gave way, or were faulty.

metal follow the contour of the section and be continuous throughout its length. Such a condition eliminates the traverse stressing of the material. Single-piece forgings are preferable

to the *built-up sections* because of the better stress distribution at the changes in section. Failures, when they do occur, however, are usually the result of critical vibrations, and the stresses developed in this manner are generally more than any metal can be expected to carry.

In order to obtain the necessary rigidity in a crankshaft, the section is large, and hence the working stresses are low. Consequently the question of strength is not such a serious matter. Bearing wear is one of the important considerations in the material and design of the shaft. Most crankshafts are made of a straight carbon steel. On the other hand, on some of the higher-speed engines, a harder material is used. In such cases, the bearings may be hardened by case carburizing or nitriding. The latter, although more expensive, is perhaps the preferable method.

Connecting rods. Connecting rods present few, if any, metallurgical problems. The majority are made of carbon steel, suitably treated. Some of the smaller high-speed engines use alloy steels to meet the higher speeds. Aluminum may be used, but as a rule, the expense is not warranted. With careful distribution of the metal in a steel rod, the section can be made so that it is light in weight, and suitable results are realized at less cost.

Bearings. In the small high-speed engines, the bearing conditions in the Diesel engine may be severe but, on the whole, they are rather normal.

As a rule, the crankshaft bearings, both main and connecting rod, are made of a thin layer of white metal backed up with a strong steel shell. The tin-base alloys, because of their greater strength, are preferred. In the fabrication of the bearings, care must be taken to see that there is a good bond between the white metal and the base and that the metal is not overheated in pouring. A *poor bond* usually results in cracking and in higher temperatures of operation. The serious disadvantage to the use of the white metals is their low-temperature operating range. When the operating temperatures are higher than the white metals can stand, the leaded bronzes are used. Their operating limit, reached through a fusion of the lead, is around 600° F. Because of the high coefficient of expansion, these metals must either be used as thin lining on a steel back, or else fitted with appreciable clearance. Seizure, when it occurs, is rather positive but usually not destructive.

The leaded bronzes are frequently used on small plain bearings and guides where the conditions of lubrication are not good. In the presence of dirt, however, they tend to load up and act as a lap. Camshaft and other accessory bearings

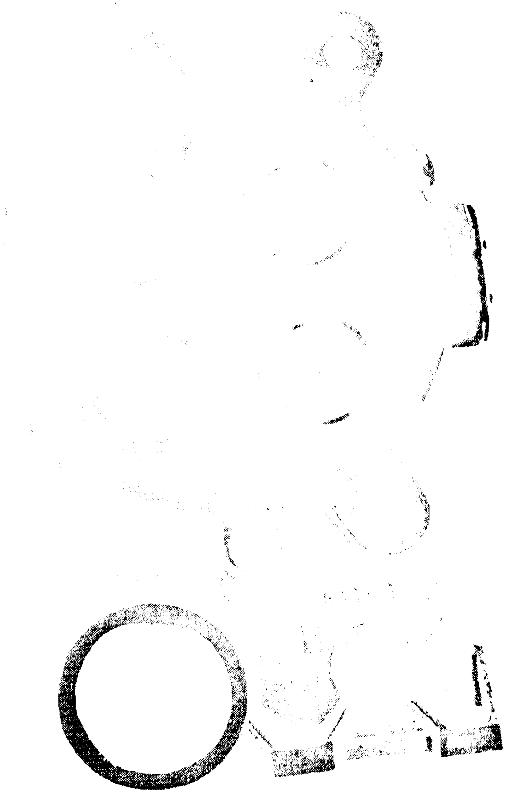


FIG. 4-11. Cylinder head, showing dual valves and air starting valve (Hendy Series 50). The head is of cast-iron construction, the metal being evenly distributed to eliminate uneven heat stresses. The intake and exhaust valve ports are of large size, scientifically designed to reduce the velocity of the air through them.

are taken care of by any one of the standard copper-tin-lead mixtures that are regularly available. Such bearings give good results when the hardness is suitably adjusted to the mating steel.

Good alignment, smooth finish, and adequate lubrication are requirements.

Valves. The exhaust valves, being perhaps the hottest spot in the engine, naturally must be made of material that will withstand the temperatures involved. In the larger engines, cast iron does very well, particularly if alloyed with chromium and nickel. In the small units where the conditions are more severe, better materials, such as chrome-nickel-silicon steels, may be necessary. Ni-Resist, stainless steel, and the chrome-nickel steels are also used. To reduce costs, a heat-resisting alloy head may be welded to a carbon steel or alloy stem.

The heat-resisting materials are rather poor conductors of heat, and the heat dissipation has, in some cases, been improved by inserting a cone of salt or copper within the head and stem. Under severe conditions, such as are found in the smaller high-speed engines, such types may serve a useful purpose.

The inlet valves, being continually cooled by the incoming air, do not require particular attention. Good results are obtained with cast iron on the larger units, and with straight alloy steels, such as a medium carbon-chrome-nickel, on the smaller engines.

The clearance between the valve stem and the valve guide must necessarily be small, and the conditions of lubrication are not good. As a result the valve stem must be made as hard as practically possible. Attempts have been made to do this through the use of chromium plating, but the results have not been overly successful. The guides are, as a rule, made of medium-hard cast iron or a bronze.

Valve seats are usually cut directly in the cylinder head material and, generally speaking, the results have very often proved satisfactory. Inserted seats of some harder or heat-resisting material may be used, but mechanical difficulties in keeping them tight make their use undesirable.

Fuel systems. Because of the high pressures involved and, at times, the corrosive nature of the oils handled, the spray nozzles and pump parts must be made of a hard, corrosion-resisting material. The stainless steels, suitably heat-treated, seem to meet the requirements. The design and construction of the different parts that are briefly discussed here will be treated in detail in later chapters.

NOTE: Metallurgical terms italicized and technical expressions related to heat treatment are defined in a glossary, Appendix I.

TABLE 4-2

COMPARATIVE PHYSICAL AND MECHANICAL PROPERTIES

Physical Constants

Material	Density lb per cu in.	Melting Range ° F	Mean Specific Heat (32°–212° F) Btu/lb/° F	Mean Coefficient of Thermal Expansion (32°–212° F) in./in./° F	Electrical Resistivity (68° F) ohms/cir.mil. ft	Thermal Conductivity (32°–212° F) Btu/sq ft/hr/° F/in.
Monel	0.319	2370–2460	.127	.0000078	290	180
Nickel	0.321	2615–2635	.13	.0000072	57	420
Inconel	0 307	2540–2600	.109	.0000064		104
"K" monel	0.306	2400–2460	.127	.0000078	350	130
"R" monel	0 319	2370–2460	.13	.0000078	290	180
"KR" monel	0.306	2400–2460	.127	.0000078	350	130
"Z" nickel	0.316	2615–2635	.13	.0000072	100	420
"D" nickel	0.317	2600	.13	.0000074	110	335
Copper	0.322	1980	.092	.0000098	10.4	2680
Cupro-nickel (70-30)	0.323	2235	.09	.0000087	220	200
Brass (yellow)	0.306	1710	.095	.0000112	40	830
Beryllium copper	0.297	1750	.1	.0000092	50	650
Nickel-silver (18%)	0.316	2030		.00001	175	230
Zinc	0.258	786	.094	see* below	36	
Wrought iron	0.278	2700–2750	.11	.0000067	87	
Carbon steel (SAE 1020)	0.284	2760	.107	.0000067	60	460
Stainless steel type 304	0.29	2550–2650	.12	.0000096	430	113
Stainless steel type 310	0.29	2550–2650	.12	.000008	480	90
Stainless steel type 316	0.29	2500–2550	.12	.0000089	440	108
Stainless steel type 330	0.28	2550–2650	.11	.0000071	600	90
Stainless steel type 430	0.28	2700–2750	.11	.0000058	360	180
Aluminum (2S)	0.098	1200–1210	.23	.0000138	17.6	1570
Aluminum (17S)	0.101	995–1190	.23	.000013	34.6	810
Lead	0.410	620	.03	.0000164	124	240

* Longitudinal test 0.000018; transverse test 0.000013.

TABLE 4-2 (*Cont.*)
Mechanical Properties

Material	Form and Condition	Yield Strength (0.20% offset) 1000 psi	Tensile Strength 1000 psi	Elongation in 2 in. per cent	Hardness Brinell	Tensile Modulus 1,000,000 psi
Monel (wrought)	Annealed	35	75	40	125	
	Hot-rolled	50	90	35	150	26
	Cold-drawn	80	100	25	190	
"R" monel	Hot-rolled	45	85	35	145	
	Cold-drawn	75	90	25	180	26
"K" monel	Hot-rolled	45	100	40	160	
	Hot-rolled[1]	110	150	25	280	26
"KR" monel	Cold-drawn	85	115	25	210	
	Cold-drawn	115	155	20	290	
Nickel (wrought)	Annealed	20	70	40	100	
	Hot-rolled	25	75	40	110	30
	Cold-drawn	70	95	25	170	
"D" nickel	Annealed	35	75	40	140	
	Hot-rolled	50	90	35	150	30
	Cold-drawn	80	100	25	190	
"Z" Nickel	Hot-rolled	50	105	35	180	
	Hot-rolled[1]	130	170	15	320	30
	Cold-drawn	90	120	25	220	
	Cold-drawn[1]	135	175	15	340	
Inconel (wrought)	Annealed	35	85	45	150	
	Hot-rolled	60	100	35	180	31
	Cold-drawn	90	115	20	200	
Cupro-nickel (70-30)	Annealed	20	50	45	80	
	Cold-drawn	50	70	20	150	
Yellow brass (high brass)	Annealed	18	45	60	60	
	Cold-rolled[2]	75	90	5	180	
Beryllium copper	Annealed	30	70	42	125	
	Cold-rolled[1,2]	97	190	2	340	
Nickel-silver (18%) (wrought)	Annealed	20	58	40	90	
	Cold-drawn[2]		105			
Carbon steel (SAE 1020)	Annealed	40	60	35	130	
	Hardened[3]	62	90	22	175	
Stainless steel type 304	Annealed	35	85	55	160	29
	Cold-rolled	100	140	20	260	
Alcoa 17S	Annealed	10	26	20	45	
	Annealed[4]	37	60	20	100	10.3
	Annealed[5]	47	65	13	101	
Copper	Annealed	10	32	40	30	
	Cold-drawn	40	45	20	105	15.8
	Cold-rolled[3]	48	52	5	120	

[1] Heat-treated. [2] Hard temper. [3] Water-quenched, drawn at 1000° F. [4] Solution heat-treated and aged. [5] Solution heat-treated, aged, and cold-worked.

TABLE 4-3*

EFFECT OF ANNEALING TEMPERATURE ON THE HARDNESS OF CAST IRON

Composition: Total Carbon, 3.42%; Graphitic Carbon, 2.95%; Combined Carbon, 0.47%; Manganese, 0.77%; Phosphorus, 0.17%, Sulphur 0.092%; Silicon, 1.87%

Specimen	Annealed 1 Hour at		Brinell Hardness Number		Scleroscope Hardness Number, Model D	Rockwell Hardness Number "B" Scale 1/16-in. Ball 100-kg Load
	° C	° F	10-mm Ball, 3000-kg Load	5-mm Ball, 750-kg Load		
No. 1	as cast	as cast	201	206	37	93
No. 2	100	212	197	196	36	91
No. 3	200	392	201	206	37	93
No. 4	300	572	197	196	36	91
No. 5	400	752	197	196	36	90
No. 6	500	932	170	178	32	85
No. 7	600	1112	114	111	25	68
No. 8	700	1292	121	121	25	71
No. 9	800	1472	156	170	30	80
No. 10	900	1652	179	187	34	85
No. 11	1000	1832	179	187	34	85

NOTE.—1/8 in. of surface material removed after heat treatment before tests were made.

"Heat Treatment Fundamentals of Plain and Alloy Cast Iron," *Metals & Alloys*, September, 1931.

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QUESTIONS

1. What alloy-steel castings are used in the manufacture of Diesel engines?

2. What are the advantages and disadvantages of steel castings?

3. Why must all steels, particularly the alloy types, be heat-treated?

* Courtesy International Nickel Co.

4. What is the principal characteristic of stainless steels as to their use?

5. What is the purpose of heat-treating forgings?

6. What advantages do the aluminum alloys possess?

7. For what use in the Diesel engine are bronzes well established?

8. Name the advantages of monel metal. In what form is it available?

9. What particular physical property should a bearing metal have?

10. What alloys are closely associated with the application of white metal?

11. When using the harder cast bronzes, what characteristics are utilized and what mechanical considerations are essential to their satisfactory performance?

12. Are the beryllium-copper alloys generally used on a wide scale as bearing metals?

13. What are the chief characteristics of chromium plating, and what advantages are claimed for it?

14. Is chromium suitable for bearing purposes?

15. Where does chromium work best, and what is its use intended to prevent?

16. To what extent are welded structures used in Diesel engines?

17. Under what conditions can cast iron be welded successfully, and what is essential to make the process successful?

18. What has welded construction to do with the weight of the engine?

19. Would it be sound practice to use welding generally to build up worn or undersize working parts?

20. What are some of the objections to the use of welding for the purpose of building up worn parts?

21. What defines a metallurgical failure of a part?

22. When does a problem of design become a metallurgical problem?

23. What is the general cause of breakage of engine parts?

24. What is the maximum stress value of a material called?

25. What causes stress concentrations, and what may they be the result of?

26. What is the prolific source of breakage in the engine parts?

27. What are the abnormal stresses due to periodic vibrations that are difficult to determine?

28. What metallurgical characteristics are related to the maintenance of a good oil film for surfaces that must be lubricated?

29. Does metal-to-metal contact occur despite lubrication?

30. When and why does oil work out of a bearing?

31. What material is usually used in the engine frame work, and what are some of the metallurgical problems?

32. To resist wear, what metallurgical characteristics should a liner have?

33. How are liners usually cast? Name the two methods.

34. Why are good cylinder head castings difficult to cast?

35. What kind of alloy may be used in cylinder head castings to prevent or retard failures through growth and fatigue.

36. What are the drawbacks of aluminum for use in making heads, and other complicated castings of that nature?

37. Why is steel not good piston material and what is the chief difficulty in using it for this purpose?

38. What metallurgical composition is used for piston rings?

39. What material is used for piston rods?

40. What composition or kind of steel is used for crankshafts?

41. What metal is used for piston pins and bushings?

42. What are the main problems of making connecting rods?

43. What kind of bearings may be made with the leaded bronzes?

44. On account of the severe operating conditions, what kind of material should be used for exhaust valves?

45. In what kind of engines may cast-iron valves be used?

CHAPTER 5

THE FUEL-MIXING PROBLEMS

THE Diesel engine differs from the gasoline engine in that the fuel in the former must be intimately mixed with the combustion air in the cylinder in a short interval of time, whereas in the gasoline engine the time necessary for the entire suction stroke and compression stroke is available for the fuel-mixing process. Gasoline fuel is made to mix readily with the intake air by its high velocity through the intake manifold and its high rate of travel through the tube of the carburetor, the heated induction passages, and the heated valve port. It is also in contact with the heated walls of the cylinder during the compression stroke. This, added to the induced turbulence during compression stroke, aids in completely vaporizing and preparing the fuel mixture for rapid ignition by the spark. In other words, this mixing process in the gasoline engine goes on over 360 deg of the crank angle, but, in the Diesel engine, the injection and mixture process must take place during the movement of the crank through only 30 to 15 deg, and hence there is but $\frac{1}{12}$ to $\frac{1}{24}$ as much time available for the mixing and preparation of the fuel for combustion in the Diesel engine as there is available for the same process in the gasoline engine.

Fuel injection methods. The systems employed to atomize and prepare the Diesel fuel for combustion during this short interval of time, particularly in the high-speed engines, must be highly perfected and extremely efficient, and maintained in this condition if anything like satisfactory operation of the Diesel engine is to be realized. The design must be suited to the type of engine and its service if the fuel is to be properly injected, finely divided, and mixed intimately with the correct amount of air to insure its rapid and complete combustion.

Diesel fuel injection systems eject the fuel at high pressures through spray nozzles into the highly heated compressed air

in the combustion space. Thus the fuel becomes thoroughly atomized, or divided into very fine particles or droplets. The combustion chamber must be of such shape that the incoming air will be given a high degree of turbulence before the fuel is injected into it. This requirement has resulted in the development of numerous designs of combustion chambers and a considerable number of arrangements of the combustion space and the locations of the fuel injection devices. It would not be possible to understand the Diesel engine without taking a number of these details into careful consideration.

Types of combustion systems. Throughout the history of the Diesel, particularly the last fifteen years, during which the high-speed engine has been brought to a high degree of development, numerous systems have been designed or proposed for the fuel injection and combustion. These systems fall into four or five general classes, any one of which may have certain features common with others, but will have a certain classification according to the method used to meet definite operating conditions and increase efficiency under those conditions.

These basic classifications predetermine the variations in engine construction and hence suggest the application to which the engine may be best suited. Obviously the various modifications of the basic system are multiplied further by the opinion of the designer, or the difference of opinion regarding such of these features as the various systems may have in common, namely, the disposition and location of the injection valves, the allotment of space in the combustion chamber, the design of the injection valve and its position with respect to the combustion chamber, the means of metering the fuel, the injection pressures employed, and many other factors according to the fancy of the designer.

These variations are more likely to be confusing to the practical operating man unless he has learned to recognize the distinctions and features that definitely place a fuel system in a designated class, and is able to see the relative importance of the various features of any system under consideration. The main classifications or distinctive features may be briefly considered from the following descriptions.

1. Open combustion chambers, called single combustion chambers. With the open combustion chamber multiple-hole spray orifices requiring very high injection pressures through the nozzle are usually associated, to atomize the fuel.

2. Precombustion chamber with direct air turbulence. Most of the precombustion chambers are distinguished by what is called a unidirectional air flow across the fuel spray, which may be injected at moderately low-injection pressure through fairly coarse single-hole injection nozzles.

3. Air-storage or air-cell chamber. Fairly low pressure is used for injection of the fuel in the air-storage or air-cell chamber, the mixing being accomplished by precombustion of a part of the fuel before it enters the main combustion space.

4. Low-compression, spark ignition fuel injection. This system employs fairly low injection pressures and divided combustion chambers. In this system the air is compressed into an air-storage chamber situated either in the cylinder head or in the piston.

Experience indicates that the open combustion chamber, known as the single combustion chamber, seems to afford the most efficient solution to the combustion and heat control problems. This system provides the highest mean effective pressures and is adaptable to small and large engines, rendering them easier to start, but the system has not been widely used in many high-speed engines. for a number of reasons that will become apparent.

In the open combustion chamber the cooled surface exposed to the combustion flame is at a minimum and hence there is less loss of heat to the cooling water, but despite its drawbacks it is the most efficient system and one with which all other systems will be compared.

The precombustion chamber has a larger cooled surface, resulting in a greater heat radiation during combustion as well as during compression. Rapid flow of heat from the combustion flame to the cooling water is thereby increased, due to concentration of the heat in the small throat and the neck communicating with the main cylinder space and due to the high velocity and temperature of the gas mixture that is projected into the cylinder for the second phase of the combustion. This feature alone restricts to a very considerable extent the efficiency of heat utilization and the mean effective pressures in the small engine cylinders.

The mean effective pressures of the precombustion chamber system are reduced further by the less efficient scavenging and lower volumetric efficiency due to a certain amount of inert gases that are usually trapped in the precombustion chamber;

moreover, the cylinder head is more complicated and heavier for this type, particularly in the 4-cycle engine. Starting the engine from "stone cold" is more difficult. Usually a higher compression is maintained to facilitate starting, further aided by heating coils or some similar device.

The advantages are obvious. Such an engine is not sensitive to the burning of the heavier, common fuel oils, and it does not require high injection pressures through multiple-hole orifices. The injection pump and nozzles will give less trouble and require less maintenance; however, a majority of the current engines are modifications of the precombustion system of fuel injection, a system that had its beginning even earlier than the Diesel engine in the hot-bulb or surface ignition type of engine.

Combustion chamber design. Combustion chamber design and development for the Diesel engine have reached such a

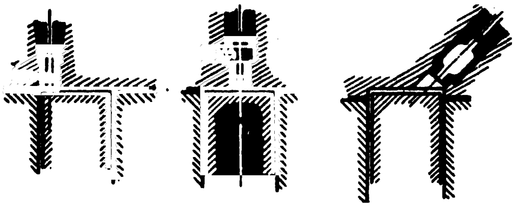


FIG. 5-1. Antechamber injection arrangements. Combustion chambers shown are of the following engines, the bore of the engine in inches being indicated: left, Deutz, 4½; center, Mercedes-Benz, 4.13; right, Hill, 3½ and 5.

point that it is no longer profitable to work out fundamentally new forms. The existing types offer a wide range for selection. The questions to consider in regard to any type are, *first*, which designs make possible good combustion at variable loads and speeds, and give high mean effective pressures; and, *second*, which designs afford these desirable results in the simplest manner.

Precombustion chamber. Combustion chambers are divided into three major groups. In the first group the fuel is injected into a small precombustion or antechamber at moderate pressures, as shown in Fig. 5-1. Some of the fuel partially burns while in the precombustion chamber, which results in a pressure rise in the precombustion chamber. This pressure rise, accompanying the first stage of combustion, projects the remaining fuel, the highly heated air, and the products of the partial combustion through the neck or throat or the holes communicating

with the main cylinder space, where the second stage of the combustion takes place automatically upon receiving additional air.

Air-storage chamber. Another class of combustion chambers mentioned is represented in Fig. 5-2, which shows the Acro system, a typical form of this class, in which the combustion air is compressed into an air-storage chamber placed in the head or the piston, as the case may be. This chamber communicates with the clearance space above the piston and under the cylinder

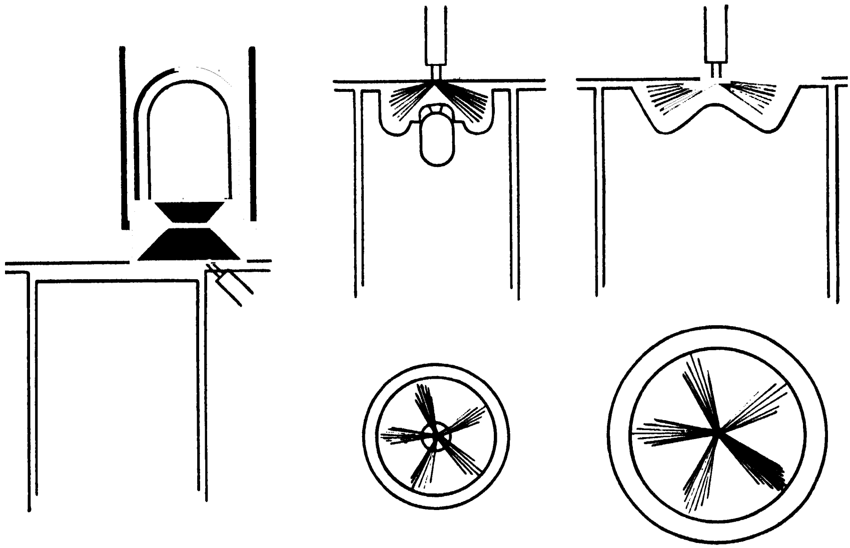


FIG. 5-2. The Acro system, Saurer engine, 4.33-in. bore. This is a French engine developed before the 1930's.

head through a narrow, funnellike passage into which the fuel is sprayed at moderately low injection pressures. The function of the air chamber is to supply air to the burning fuel, making the combustion process somewhat similar to that in a blow torch. This type of chamber and the general designs of the antechamber have the advantage of low fuel pump pressure and coarse jets through single-hole orifices as a rule and the disadvantage of being hard to start and having lower thermal efficiency.

Single combustion chambers and their efficiency. The third class of Diesel combustion chambers comprises the single combustion chamber into which the fuel is injected at high pressures

either through one central injection valve with a single- or multiple-hole nozzle or through one or two injection valves from the side. Examples of this class are shown in Figs. 5-3, 5-4, and

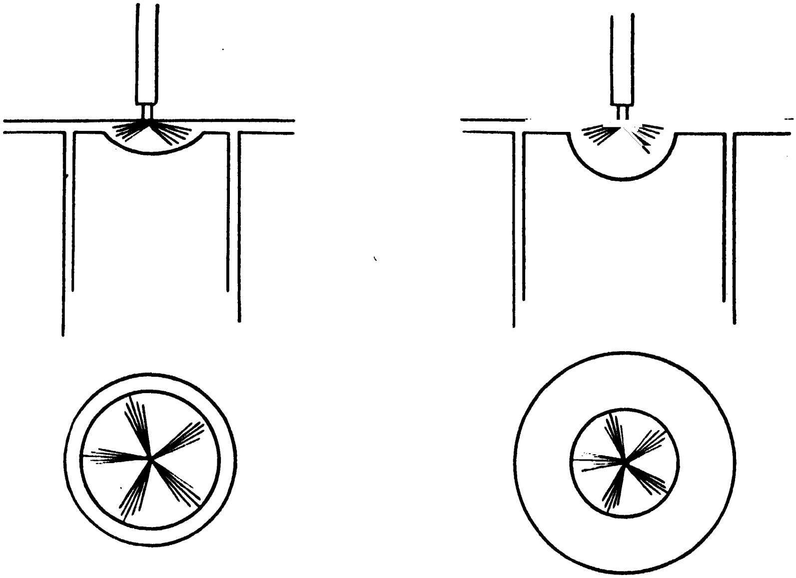


FIG. 5-3. Single combustion chambers with central injection.

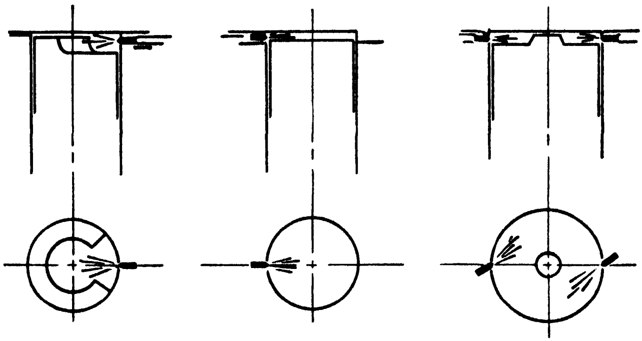


FIG. 5-4. Single combustion chambers with side injection. These were the early forms originated prior to 1930.

5-5. It should be noted that the examples of this type of chamber indicate great differences in regard to shape. Usually the form of the combustion chamber is such that the shape is adapted to the direction and form of the spray jet. The air

intake valves and their ports are arranged so as to promote directed air turbulence, the piston causing additional air whirl as it approaches upper dead center. This results in giving the whole mass of rotating and whirling air a movement that makes possible the penetration of the fuel jets.

Considerable attention has been devoted to the development of single combustion chambers in which the combustion is aided by directed air turbulence as a direct method of attaining high mean effective pressures with low fuel consumption with wide ranges of load and speed. When fine spray atomization

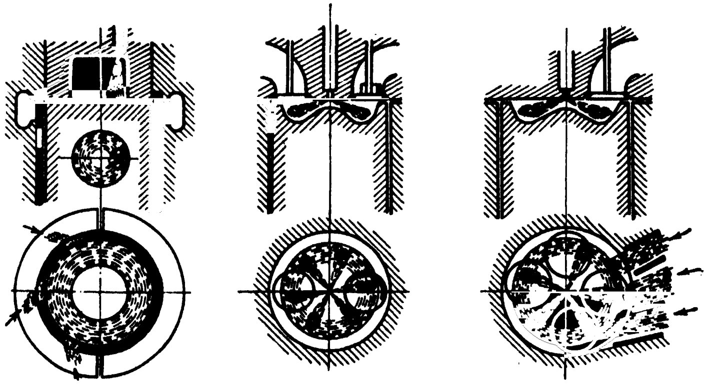
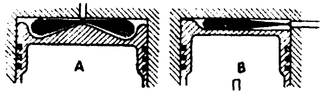


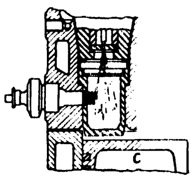
FIG. 5-5. Single combustion chambers with directed turbulence and central injection. The sketch at the left represents Ricardo engines of $5\frac{1}{8}$-in. and $7\frac{1}{2}$-in. bores. The central sketch is of the Hesselman engine, and the sketch at the right is of a 9-in. bore engine. This engine was developed around the early 1930's.

by high-pressure injection is employed to obtain this result, the nozzle holes must be very small and numerous, and exceedingly high pump pressures for the larger cylinders are needed, a requirement that imposes a number of disadvantages.

Uncontrolled turbulence, caused by the quick displacement of the air entrapped between the outer part of the piston top and the bottom of the cylinder head, may be developed in any of the single combustion chambers shown in Figs. 5-3, 5-4, and 5-5. The wider the flat piston rim and the smaller the piston end clearance, the greater is the turbulence. The width of the rim of a dished piston is limited by the necessity of making room for the spray. The rim is usually wide for large cylinders, but it is necessarily narrow in the case of small engines to avoid the necessity for very fine spray holes in the orifices of the nozzle.

Controlled turbulence in conjunction with central injection has been applied by Hesselman. His design included a shrouded portion of the inlet valve, as shown in Fig. 5-5, so that the air flows into the cylinder tangentially and rotates, after being compressed, at such speeds that it is made to move from one fuel jet to the next during the injection period. Later, we shall see how Ricardo made use of this principle in his high-speed, sleeve valve engine, illustrated diagrammatically in Fig. 5-5. The test results that were reported on this engine were very interesting. The indicated mean effective pressure at the normal speed of 1300 rpm was given as 112 psi with a fuel consumption of 0.29 lb per ihp; and 122 psi was the indicated mean





CLASSIFICATION CHART

Injection and Combustion Methods

Methods - Diagram	A	B	C	D	E	F
Single Chamber	✓	✓			✓	✓
Pre-combustion			✓	✓		
Turbulence type	✓	✓				
Non-turbulence					✓	✓
Side Injection		✓				
Central Injection	✓		✓	✓	✓	✓
Single Orifice			✓	✓		
Multiple Orifice	✓				✓	✓

FIG. 5-6. Classification of combustion chambers.

effective pressure with a fuel consumption of 0.285 pounds per ihp at 2000 rpm. A fuel consumption of 0.38 lb per bhp was guaranteed for the $7\frac{1}{2} \times 12$ in. Ricardo engine at 900 rpm. This was a remarkable result.

Ricardo's results were obtained with a single solid fuel jet from one large open nozzle of $\frac{1}{32}$-in. diameter, placed vertically, near the high cylindrical wall of the combustion chamber as shown in Fig. 5-5. Tangential ports in the sleeve valve direct the air into the cylinder in a circular current, the angular speed of which is approximately doubled when the air stream is transferred, without losing peripheral speed, into the compression space having a diameter about one half that of the main cylinder. Uncontrolled turbulence is superimposed upon the directed air turbulence or rotational swirl near top dead center by the displacement of the piston. Ricardo found by means of numerous experiments that the best results are obtainable when the incoming air is so directed that it makes one revolution within the combustion chamber in a crank angle of

from 30 to 36 deg, which is identical with the injection period at full load.

An attractive feature of Ricardo's high turbulence chamber is the simple spray nozzle having a hole of unusually large size. This feature eliminates the danger of choked nozzles and also allows low fuel injection pump pressures. This is desirable as high pressures mean increased maintenance cost of the fuel injection pump parts. The advantages enumerated are, however, offset by the necessity for a sleeve valve construction, which leads to expensive mechanical details and complications.

Directed turbulence effect has been tried in a number of designs in recent years, such as the arrangement of simple dual

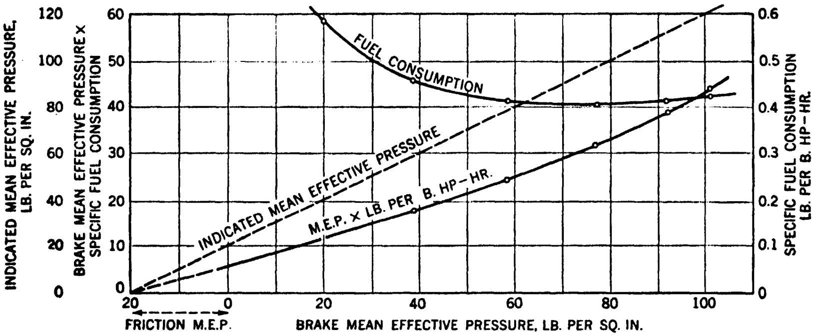


FIG. 5-7. Test curves of 8½ × 12-in. two-cylinder experimental engine. Ontario crude oil was used as fuel, the engine running at 600 rpm. Examples of the use of curves such as this are given in a paper by Otto Nonnenbruch.

inlet valves represented diagrammatically in Fig. 5-5. The valves were placed and the inlet ports so shaped that the air could be drawn in spirally and thus induce a rotational motion on the vortex-shaped piston, and the circular motion of the air was intensified during the compression stroke. The fuel is injected through a multiple-orifice spray valve, centrally located so as to direct the jets close to the piston but not to touch it, with the result that the air immediately below the flat cylinder head expands through the burning fuel jets during the down stroke of the piston. It was desired that turbulence of the air charge would take place at the desired speed, which would increase or decrease in proportion to the speed of the engine. This experimental engine afforded an opportunity to study the various arrangements and reports made to the ASME were very enlightening.

Desirable factors for economical operation of high-speed engines. In order to maintain the economical operation of the Diesel engine, it is concluded that certain desirable results must be realized in practice for any system of combustion. To enumerate these factors:

1. Minimum fuel system maintenance with maximum reliability obtainable by the use of a single-hole orifice channeled to a single-plunger fuel pump for each cylinder has been the desideratum.
2. Colorless exhaust at all operating loads and speeds made possible by sustained pressure fuel atomization with positively controlled start of the injection period and sharp cutoff.
3. Individual measuring type of fuel pumps for close regulation of minute fuel quantities to assure accurate distribution, obtainable by injection to the cylinders of a finely atomized spray for satisfactory idling and slow-speed operation.
4. Rapid heat transformation of the injected fuel by good distribution and correct and sufficient turbulence obtainable with high temperatures.
5. Reasonably low maximum pressures reached at limited accelerated rates of pressure rise in the combustion chamber.

Comparative conclusions. To continue the comparison of the two major kinds of precombustion chambers and the shapes that aim at the results enumerated, the two types may be studied as shown in Fig. 5-8. These two types are designed for the

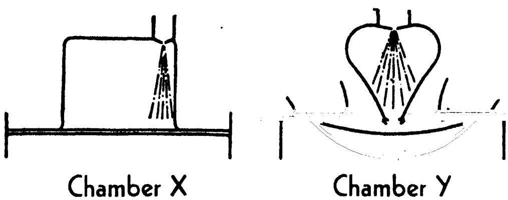


FIG. 5-8. Two types of combustion chambers designed to control combustion. Chamber X regulates rate of fuel feed, while chamber Y is a precombustion chamber of limited oxygen content. Chamber X gives lower fuel consumption, while chamber Y gives greater smoothness and permits higher brake mean effective pressures.

control of the combustion in the manner and for the purpose indicated, but they differ in shape and design. Chamber X regulates the rate of the fuel feed, and chamber Y has a precombustion chamber, or antechamber, of limited capacity for air. The quality of combustion can be made all that is desired in

either type of chamber, but chamber X gives a lower fuel consumption and a higher brake mean effective pressure, whereas chamber Y gives a greater degree of smoothness in combustion and permits high sustained brake mean effective pressures. The question may be asked then, Why the difference?

The shape of the fuel consumption curve has greater significance to the operating man than has the actual minimum figure of fuel consumption obtained at certain loads and conditions. A close approach to the flat parabolic form of curve is highly desirable, whereas in practice it is frequently the case that the hyperbolic form of curve is found. In other words, the Diesel engine, if it is to be highly successful, should not have a high fuel consumption at light loads and idling speeds. This will be the case unless the fuel injection and combustion systems are adapted to the application and the design of the engine. Combustion system X is lower in fuel consumption simply because it will have *lower heat losses* than chamber Y, particularly at lower loads and idling speeds, because it has a *compact combustion* chamber with less surface exposed. Chamber Y has a greater smoothness of combustion than chamber X and therefore will permit a higher sustained *brake mean* effective pressure. In practice the two types are approximately balanced in output.

Combustion smoothness is of great importance if it can be obtained by simple automatic means. Volume control of the burning gases, accomplished in chamber Y, has a more practical appeal to the operator than has the fuel volume control in chamber X. In this connection, it is evident that a practical compromise must be made between low fuel consumption and smoothness of operation, between brake mean effective pressure and low maintenance cost, between ideal full-load operating economy and wide flexibility of speed with fair fuel economy at all speeds.

Good idling is essential in applications such as those in mobile equipment. In such applications all cylinders should be able to contribute equally in providing part of the load for uniform idling, even for extended periods. An engine with the Y-type chamber will idle to lower speeds with positive ignition than will one with the X-type chamber. Ignition of the preliminary fuel charge is the same regardless of the load or speed with the Y-type chamber, the fuel consumption and thermal conditions remaining practically constant over the entire range of operating speeds and loads.

Starting is easier, however, with the compact chambers of the X-type, as the more concentrated and less divided the combustion space, the lower will be the compression pressures necessary for starting the engine when cold. System X, therefore, has superior *starting* characteristics compared with Y. The latter system usually requires electric heating coils to assist in starting in cold weather.

There are many modifications of each design. It may be assumed that directed air turbulence in open chambers or the use of precombustion chambers in combination with various fuel injection nozzles is common to the average high-speed engine. The nozzles for injection of the fuel are either centrally placed in the top of the cylinder head, or placed at the side of the cylinder head, depending on the shape and design of the combustion chamber.

The common-rail system of fuel injection, for example, may be used for the various combustion chambers shown in Fig. 5-4. On the other hand, an individual pump for each cylinder may be used for the various combustion chambers, a method that is used with the high-speed Diesel engine. Exception to this is the use of a single-plunger pump and a distributor disc, functioning similarly to the distributor on the ignition system of the gasoline engine. The shape, size, function, location, and design of the combustion chamber, in conjunction with a predetermined fuel injection process, have occupied the attention of designers and inventors for forty years. The United States Patent Office has issued more than 1000 modifications of the Diesel combustion chamber, and it frequently appears that a greater number are in use than are justified. It can be said that despite this great number of patents and designs, they are only variations of the few basic types here discussed. Only a small number are in practical everyday commercial use.

Turbulence, atomization, and vaporization. In the Diesel engine, ignition of the injected fuel immediately following its entrance into the combustion space is desired. If rapid and instantaneous ignition is to be obtained, the fuel must reach its ignition temperature in an interval of time so short as to be measured only in thousandths of a second, as already shown. The fuel must be injected into an atmosphere the temperature of which is higher than the ignition temperature of the fuel. This temperature is obtained by means of high compression.

Fuel oil is difficult to ignite, and therefore the air temperature in the combustion chamber must be higher than in the gasoline engine to generate sufficient heat to fire the fuel charge. In the carburetor engine, on the other hand, the fuel is introduced into the cylinder during the intake stroke. In order to avoid spontaneous and uncontrolled ignition in the gasoline engine, it is necessary to keep the temperature and, consequently, the compression ratio relatively low. In the carburetor engine, the heating of the fuel is objectionable, but in Diesel engines it is *very desirable.*

Fuel mixing in the Diesel engine. Obviously the heating of the fuel is a question of temperature, but it must not be overlooked that it is also always a question of time. In a carburetor engine the fuel picks up heat from the walls of the combustion chamber, the fuel being exposed to the heat of compression during the whole of the compression stroke. The factor that limits the output of an engine of a given size is the air it will pump. It is possible to supply almost an unlimited quantity of fuel to an engine, but the amount of the air supply to the cylinder is limited by the swept volume, unless supercharged. It is necessary therefore to make the best possible use of the air. In the carburetor engine, the fuel is supplied to the cylinder during the suction stroke and there is sufficient time to mix the fuel with the air, evaporating or vaporizing it thoroughly, so that at the end of the compression stroke there is a homogeneous gas mixture ready to be ignited. When the spark ignites this mixture, one of the most efficient processes of combustion known to science occurs. In the Diesel engine, however, the mixing of the fuel with air is rather a difficult problem. Much of the development work on the modern Diesel engine has been concentrated on the solution of this problem. In fact, it is two problems, and it is the purpose here to show how the designer of engines sets about to accomplish the solutions to the fuel-mixing problem in the Diesel engine. A restatement of the fundamentals shows just what the two problems involve:

1. The fuel must be measured accurately and evenly over long periods of operation, and injected in small quantities into the various cylinders during their working stroke, with as great reliability and flexibility of range of operation as possible.

2. The fuel must be prepared before injection, so that when it is injected, the charge enters the combustion chamber already prepared so far as mechanically possible to mix with the air for

rapid and complete combustion; otherwise, it is not possible to obtain perfect ignition and burning of the fuel in the Diesel engine. It is necessary that the injected fuel be evenly distributed throughout the whole air charge in the combustion space as quickly as possible. In order to obtain this distribution designers have formed the combustion chamber so as to suit the shape of the spray. The air charge has been given a rotational, whirling, or turbulent motion, requiring the use of various kinds of combustion chambers.

It is a well-known fact, however, that the Diesel engine does not utilize the air very successfully, or efficiently, and for most designs, it is necessary to supply a considerable amount of excess air. There are two factors that control the mixing of the fuel with the air. One is space, the other is the time available for the mixing process. The first factor, space, is usually well taken care of in most engines. The matter of time, however, cannot be taken care of so easily, for in the Diesel engine, it is necessary that the fuel be burned immediately or as soon as possible after it comes in contact with the air in the combustion chamber.

Only the first fuel injected comes in contact with absolutely fresh air. The fuel thereafter has to be injected through air and also products of combustion where the air has already been utilized. This brings about a condition wherein some parts of the combustion chamber contain more fuel in a given space than can be burned without smoke, owing to insufficient oxygen, whereas in other parts of the chamber there is a surplus of air that cannot be reached by the fuel and utilized, except by the use of multiple-hole nozzles. To solve this problem, designers, instead of bringing the fuel to the air, bring the air to the fuel by various methods by which a whirling motion is imparted to the air flow, creating turbulence, as already shown.

Object of designs of combustion chambers. The object of most designs of combustion chambers is to create turbulence, so easily attained in the air injection engine having a simple disc-type combustion space, and to bring about the high degree of vaporization attained in the air injection engine. When this degree of vaporization can be realized by the design of the combustion chamber or by any method of preparing the fuel for ignition, rather than depending upon extremely high injection pressures, the designer has attained a great measure of success with his design.

The result of turbulence and fuel-mixing methods is the increase of the rate of heat transfer between the air in the chamber and the atomized fuel by virtue of the violent agitation that promotes more intimate mixing of the cylinder contents during the injection and ignition periods. Either turbulence or high injection pressure through multiple-hole nozzles is required to prevent a fuel fog from stagnating around the tip of the spray orifice, thus leaving the remaining parts of the combustion air without fuel saturation; a condition that retards the speed of combustion or flame propagation, which should take place throughout the combustion chamber uniformly. This delay of the combustion process produces *after-burning* in the Diesel, and lowers to a great extent the effective capacity of the engine. Maintenance trouble and high repair cost may result; hence after-burning must be prevented if the engine is to be commercially successful.

The principal method of bringing about a more intimate mixture of fuel and air is the use of the precombustion chamber, or a *preignition chamber.* In the chamber, fuel is injected into the space inserted in the cylinder head, from which it is blown into the main combustion space, directly over the piston, by initial pressure caused by partial combustion of the fuel before its entrance into the main combustion chamber. The space in some precombustion chambers is closed almost entirely by being connected with the main cylinder space by one or more small holes. Many precombustion chambers of this type consist of a cavity, usually in the cylinder head, sometimes in the cylinder, having a volume approximately 30 per cent of the total clearance volume. The advancing piston forces the air into this chamber, where it meets the oil spraying in from the fuel injector. The heat vaporizes the fuel, mixes it with the air, and ignites it, whereupon complete ignition occurs when it is forced out into the main combustion space above the piston into sufficient additional air to complete the process. These various devices for preparing the fuel charge before ignition are different in purpose and are called turbulence chambers, preignition chambers, chambers with air cell, combined turbulence and precombustion chambers, and so on.

The preignition chamber has several functions:

1. To foster and direct the initiating of ignition.
2. To distribute and mix the fuel throughout the combustion air.

3. To aid in putting the fuel in proper combustible form or condition by warming it, or supplying hot air, and thus aiding combustion and producing vaporization. The first two functions are fulfilled in all types of chambers of this class; the third function is performed by means of a special combination of the air chamber and the method of injection of the fuel, as will be explained.

In one type of precombustion chamber, the fuel is fed into the neck of the chamber, the chief function being that of fuel atomization and distribution. The air, trapped in the antechamber during compression together with gases from a small amount of combustion that occurs within the chamber neck, serves to spray the fuel injected at the midpoint of the chamber, similar to the effect produced by an air injection blast. Where the usual antechamber serves as a preignition or precombustion chamber, all the fuel is made to pass through the chamber. When the chamber is small and contains only sufficient air to burn part of the fuel injected into it, the resulting partial combustion blows the fuel out of the antechamber through holes or grids. This serves to produce rapid flame distribution of the unburned fuel throughout the main combustion space in a manner similar to that of a carburetor engine. The more complete this vaporization in the precombustion chamber or air cell, the more successful and complete the final combustion that follows.

When the antechamber is large and serves as a vaporizer and ignition chamber for most of the fuel, it is essentially akin to the early designs known as the "hot-bulb" engine. In this case the antechamber has a large neck and discharges a large mass of flame, instead of a small mass as in the small-neck type, or small streams of flame as in the multiple-orifice type.

Thus it is evident that the main problem, particularly with high-speed engines, is to provide ample time for injection, formation of an inflammable mixture, and ignition before the piston has moved too far on its out-stroke, and without having ignition occur too early on the compression stroke. The greater the variation in rpm and the load, the more involved the problem. The time required for making the combustible mixture and the *time lag of ignition* depend in part upon the available heat and its temperature. It has been found by many designers to be desirable to create what is called a *pilot quantity* of fuel vaporization in order to reduce the time lag of ignition. This fuel

which makes possible much lower pressures without sacrificing good combustion, power, and economy.

The fuel, leaving the nozzle and passing along the common center of two lobes, the latter formed under both valves, inlet and exhaust, and directly over the cylinder proper, passes through the hot compressed air that is generated in the compression stroke, into major and minor air chambers, or what are called *energy chambers.* The fuel mixture, fuel in conjunction with air picked up in the passage of the fuel through a venturi-type orifice, passes through the orifice and causes violent combustion upon its entry into the energy chambers. The air-

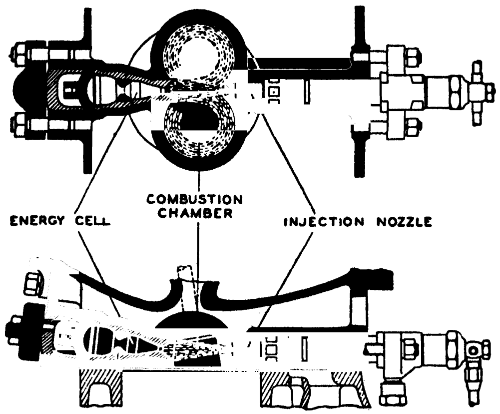


FIG. 5-11. Schematic drawing of the Mack-Lanova combustion system.

fuel mixture that is ignited in these energy chambers discharges back violently into the main chamber or combustion space above the piston, thus impinging on such residual fuel sprays as are still leaving the nozzle or are in suspension. This counter-flow, backfire, or reversal of the contents of the combustion chamber sets up a violent turbulence, which is divided in such a manner as to create the right and left rotary motion in the lobe under each respective valve. This brings about what is called *self-induced turbulence,* which thoroughly mixes all atomized fuel with its proper quantity of air. Combustion, being already under way, proceeds in the desired manner. This system of induced turbulence by internal means gives what has been long sought for in the solid injection engine, a substitute for air injection effect. It also produces accelerated burning and progressive rise in pressure by accelerating the end of the com-

bustion cycle instead of the beginning and thus more closely approaches the characteristics of the gasoline engine.

The air-cell and organized turbulence. Another method of rotating violently the combustion air to obtain complete

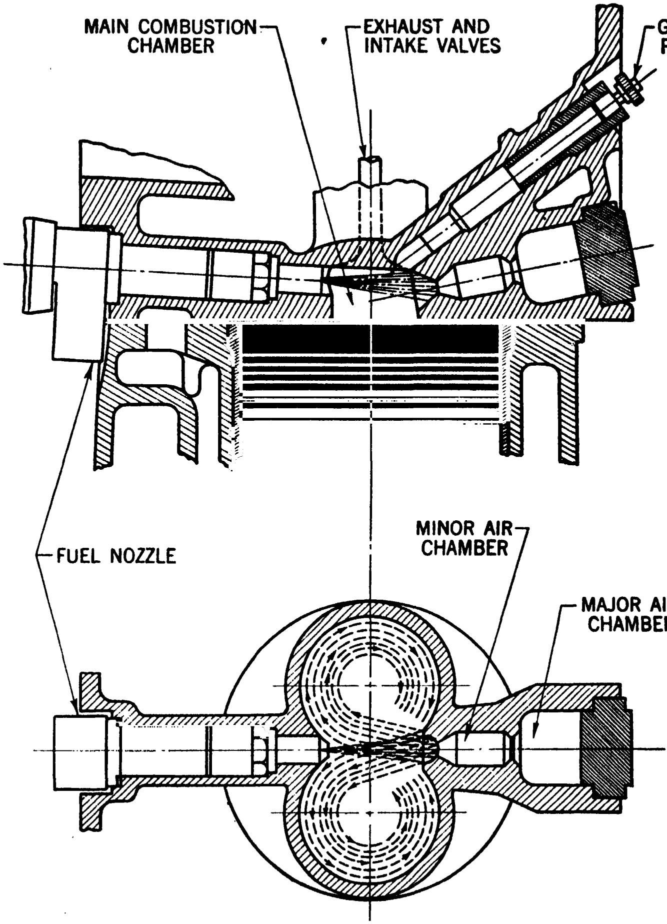


Fig. 5-12. An early design of the Lanova system.

combustion, called *organized combustion,* is shown in Figs. 5-13 and 5-14. The combustion chamber is so designed that the throat connecting the spherical combustion chamber with the cylinder is situated on one side of the sphere. As a result of combustion there is a natural rise in pressure in the cylinder, forcing the piston to descend on its working stroke. The

combustion chamber of the *Comet* engine, designed by Ricardo for the purpose of eliminating Diesel knock, employs this organized turbulence method by the use of the air cell. The chamber is of two-piece construction. The upper half of the cup is integral with the head; the lower half is of Hoskins metal and is held in place by a loosely threaded bronze lock ring. This construction provides for good heat regulation at the cup and around the water-jacketed injector nozzle, and makes it possible to inspect the inside of the combustion chamber, as well as permitting the chamber throat to be operated at higher temperatures than otherwise would be feasible.

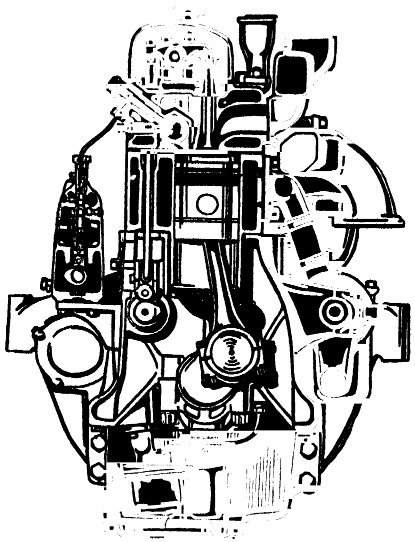


FIG. 5-13. Century Comet Six (Waukesha) showing non-rocking, truncated cylinders, five-ring aluminum piston, single-piece combustion chamber insert, pintle-type injector, and glow plug for cold-weather starting. More than 85 per cent of the intake air is compressed into the combustion chamber. This is after the Ricardo design.

The outside appearance of the combustion chamber is shown to be cylindrical and to have a flanged bottom. The inside of the combustion chamber is a spherical air cell. This is approached tangentially by a venturi-shaped throat that connects the engine cylinder with the combustion chamber. The working principles are as follows:

1. The piston coming up on its compression stroke forces the air through the throat and into the spherical air cell where it is compressed to a pressure of approximately 550 psi. The clearance between the top of the piston and the cylinder head is between 0.040 and 0.050 in.; therefore all the air in the cylinder except in this very narrow clearance space is forced into the relatively small combustion chamber and the throat leading to it.

2. The tangential relation between the throat and the combustion chamber forces the air to *swirl* around the vertical axis of the combustion chamber; and, passing through the injector in this manner, the air sweeps the injector at all times. The

heat loss through the tangential port due to high velocity air flow is reduced to some extent by fitting a heat-insulating *hot plug*. This contains not only the tangential port, but also the lower half of the spherical chamber. This hot plug also assists in reducing the delay period, thus giving smoother combustion.

3. The high-speed swirling action of the air is the result of the chamber design. This swirling action makes sure that the injected fuel is mixed immediately and intimately with the air, permitting high-speed operation, clear exhaust, and high power output. The speed of the swirling air is in direct proportion or relation to the speed of the engine, which, in itself, is a governing device for the period of combustion. The antechamber is fitted with electric glow plugs for starting purposes when cold.

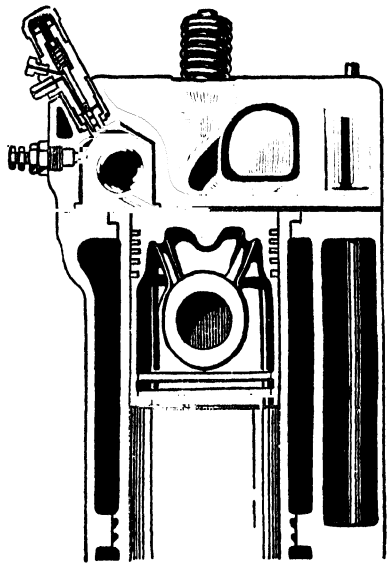


Fig. 5-14. Silver Comet Cylinder and combustion chamber in section view, showing the wet-sleeve liner, five-ring aluminum piston, single-piece combustion chamber insert, pintle-type injector, and glow-plug for cold-weather starting.

Vaporizing precombustion chambers. The vaporizing precombustion or preignition chamber gives a violent turbulence and intimate mixture of fuel and air in both the antechamber and cylinder. The engine thus works efficiently with a weak air mixture, that is, with an excess of air, and the exhaust is smokeless.

Positive ignition at all loads and speeds is accomplished by always burning the same amount of fuel in the combustion chamber and thus maintaining a high temperature. The radiating surface is so limited that an adequate temperature is available at all times for positive ignition. It provides for preparing the fuel for final combustion by partially burning it in the precombustion chamber. When the fuel injection valve sprays the fuel into the precombustion chamber in the form of a mist, the highly heated air causes a small portion of the fuel to ignite. As the injection continues, the fuel is enveloped by

flame, the entire mass becomes gaseous, and, by virtue of the high pressures developed, rushes at high velocity into the main cylinder, where further combustion occurs. Expansion then takes place and the rest of the fuel injected is changed into a rich, *dry mixture*, passing on into the main combustion chamber at high velocity. A high turbulence is created in the cylinder combustion space, causing the gaseous fuel to become thoroughly mixed with the oxygen from the incoming air, the condition desired for complete combustion.

The antechamber can be so designed as to have a higher temperature than could be allowed for the main combustion chamber and cylinder walls. For variable loads, the volume of the combustion chamber of the antechamber type can be designed for the combustion of a somewhat constant volume of fuel, which tends to maintain a constant temperature. This is accomplished by making the volume of the antechamber smaller, small enough so that at idling speeds all the fuel charge will not be burned, because of lack of oxygen. Heating and cooling depend on time and, with a variable speed, care is necessary to control the temperature.

Positive pressure precombustion chambers. Instead of using the pressure rise, resulting from partial initial combustion within the antechamber or air cell, to force the fuel out, another type of chamber (Cummins) employs cam-actuated plungers to displace the fuel from the chamber into the cylinder space. The plunger is described as moving outward during the suction stroke of the engine, the fuel being sucked into the small space under the end of the plunger, and thus the fuel, air, and vapor being forced into the main cylinder space. Two conical cups are used with the plunger seated on the inner cup by the action of the cam. A fluted projection on the inner cup causes a thin space to be left between the two cups, the fuel being delivered to this space through small spring-loaded ball check valves. The fuel is then sucked into the flutes by the upward movement of the plunger. The backflow or return of fuel during the injection stroke is prevented by a check valve. The fuel is ejected into the cylinder through fine orifices drilled in the outer cup. It is evident that the thin walls are subject to high temperature, with the fuel remaining in the cups a considerable time, which condition undoubtedly results in vaporization while the fuel is under the plunger. This plunger also sucks in some air from the working cylinder during the compression stroke of

the piston. When injection begins, the liquid fuel standing in the lower part of the cup is the first to be expelled, while at the end of the process, the discharge of air and vapor evidently sweeps the orifices dry and protects them from incrustation of

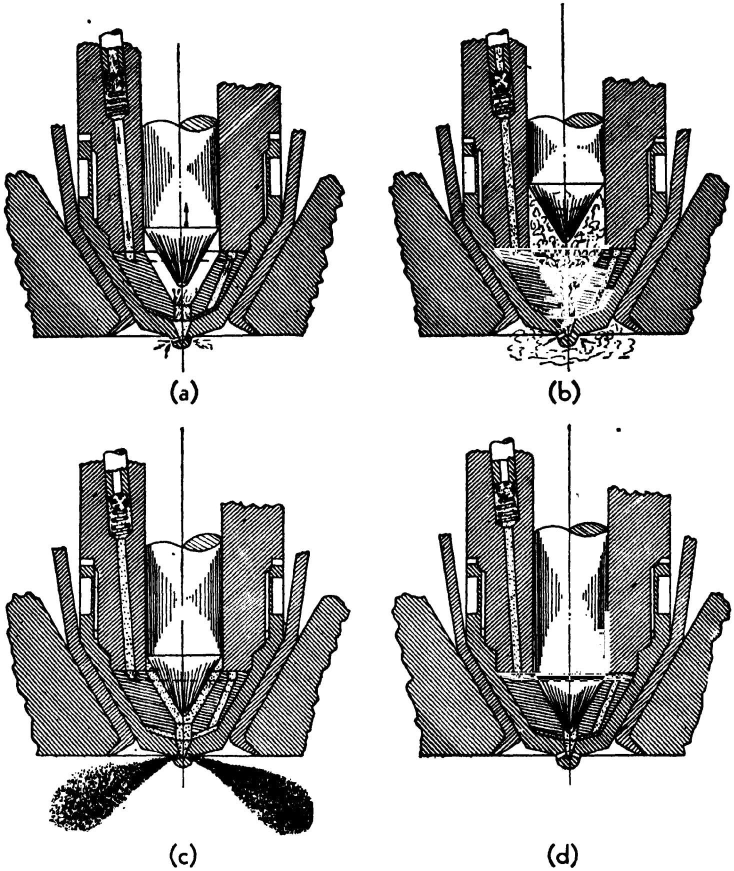


FIG. 5-15. Cummins Diesel injector operation. (a) Fuel check valve open. Plunger moving up and fuel entering plunger chamber. (b) Plunger at top position. Check valve closed. Correct amount of fuel in plunger chamber, hot air from compression being forced through fuel charge. (c) Plunger moving down, driving gasified fuel charge into combustion chamber. (d) Plunger seated and injection finished.

carbon that would result if the oil were not completely dispelled at each stroke.

As shown in Fig. 5-15, after the pump has measured the proper fuel charge and delivered it to the injectors, the fuel is

preheated or warmed by circulating through the preheating chambers in the end of the injector before entering the precombustion chamber. It is then forced into the injector at a point directly above the nozzle, and on the compression stroke it is further gasified. Then, with the piston near top center, the plunger is suddenly depressed by the proper cam action, driving this preheated charge of gaseous vapor back through the nozzle into the combustion chamber, where ignition takes place.

Mechanism of fuel systems and preparation of fuel for combustion. The principal methods of combustion and the way by which the fuel is mixed with the air in the combustion chamber by means of the design of the combustion chamber and the air intake system having been outlined, a study will now be made of the combustion process and how it is brought about by means of the spray nozzle and fuel pump acting together as a high-pressure hydraulic unit, and of its function of delivering the fuel from the supply tank directly into the engine cylinder.

It has been shown how the fuel is thoroughly atomized as it is projected into the air charge, which has already attained a temperature above that of the fuel's ignition point, at which it is ignited and burned according to the Diesel principle of combustion. Upon injection into this air, each globule of fuel at once absorbs heat from the surrounding air. These minute particles of fuel, traveling through the dense compressed air charge, begin to *vaporize* as soon as the boiling point is reached. When the compression temperature is sufficiently above the ignition point of the fuel, the vapors thus formed will ignite immediately and the fuel charge burn uniformly, the ignition commencing from the *surface of the droplets* as vapor is formed.

In the event that the compression temperature is too low for instantaneous ignition of the fuel vapor, the fuel will continue to accumulate until the ignition eventually occurs.

Temperatures and pressures during injection. A diagram of pressures and temperatures occurring during the injection process is shown in Fig. 5-16. When the fuel vapor mixes with the air under the conditions outlined and forms the combustible mixture, it will burn at a faster rate when it eventually ignites than if the ignition began sooner. In the latter event there is a strong tendency for the combustion of the entire fuel charge to occur spontaneously throughout the whole of the combustion chamber, with the result that the pressure rise will be very rapid.

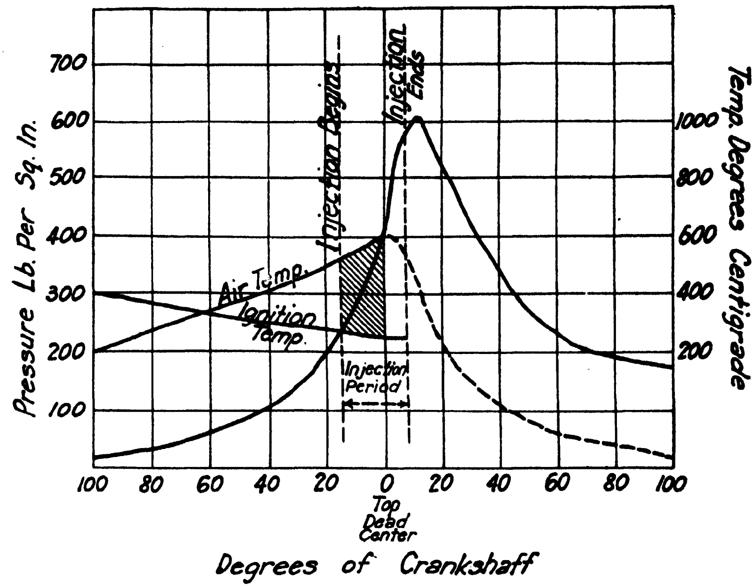


FIG. 5-16. Pressure related to the injection process and temperatures developed.

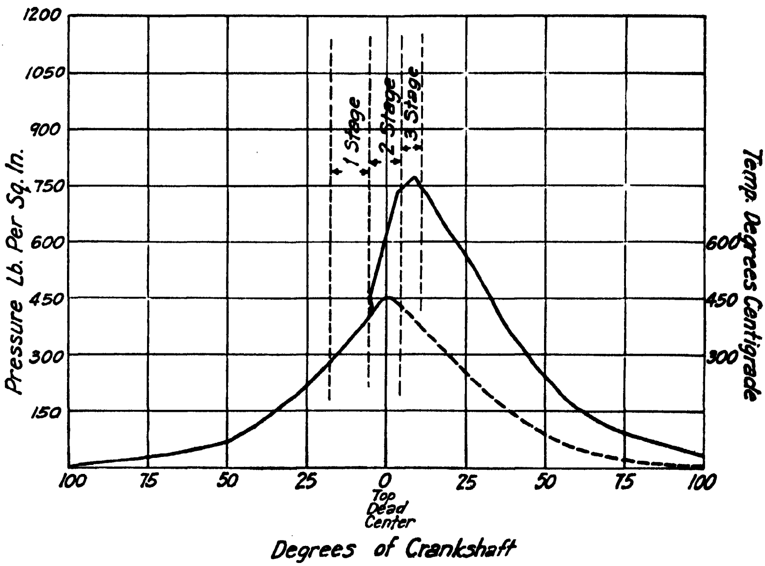


FIG. 5-17. The three stages of combustion in relation to the pressure and temperature developed.

When this pressure rise is excessive in value, a very definite detonation occurs, which is called "Diesel knock."

Three stages of combustion. Actual indicator cards have been studied by such well-known designers as Ricardo of England, who suggested an explanation of the combustion process that is now generally accepted in principle. He defined the combustion process as consisting of three definite and distinct stages during the period of fuel injection. Visualizing the conditions occurring in the cylinder during this interval as indicated in Fig. 5-16, we can examine Fig. 5-17, designed to illustrate the *three stages of combustion.*

1. First stage of combustion. The first stage of combustion is known as an initial delay period and referred to as *ignition lag* in Diesel engine literature. It is during this period that fuel is being injected, but no ignition occurs. The length of this period is extremely important, and the shorter it can be made, the more satisfactory is the operation of the engine. Ignition does not begin until some minute portions of the fuel droplets are vaporized and mixed with sufficient oxygen to form a combustible mixture, and the temperature must be sufficient also. It is supposed that, once the ignition commences, the flame spreads from the nuclear points in a manner similar to that in the spark ignition engine.

This delay period is constant in terms of time rather than degrees of crank angle, but a great deal depends upon the chemical nature of the fuel, the pressure of the highly compressed air, and the degree of atomization of the fuel as it mixes with the air. Temperatures, injection pressures, and the degree of turbulence attained by the air during the injection period have a marked influence on ignition lag. In general, it is intended that the fuel should begin to burn as it arrives; while this is almost impossible to expect, engineers have by various means been able to shorten the period of ignition lag.

2. Second stage of combustion. The second stage of combustion is the period of rapid ignition that results from the spontaneous spread of the flame (the beginning of which ended the first stage) to the main area of the combustion space. As is the case with the gasoline engine, this rate of flame spread and, consequently, the rate of pressure rise are dependent upon turbulence and other factors. This rate of pressure rise is constant in terms of *crank angle* rather than time as is true of ignition lag, or the first phase of injection.

The second phase of combustion has been modified considerably by turbulence, as already pointed out, and in this respect differs significantly from the gasoline engine, for, in the gasoline engine, *all the fuel* is in the combustion chamber already mixed with the air, whereas in the Diesel engine only a portion of the fuel so far has been injected and prepared for ignition by mixing. It is for this reason that the pressure, although its rate of rise is of the same order as that of the gasoline engine with similar turbulence, does not attain anything like the same maximum value it would were sufficient fuel present in the combustion space at this time to combine with all the oxygen in the charge.

For this reason maximum combustion pressures in the Diesel engine should be relatively higher than in the gasoline engine, and it can now be understood that the pressure rise reached at the end of this second phase of the combustion process depends upon the duration of the *delay period*, or ignition lag, which is influenced by engine speed, the rate of injection, the timing of injection, and finally upon both the temperature and pressure of the compressed air charge.

3. Third stage of combustion. The third stage of combustion is a period during which the remaining fuel is injected and burns as it enters the combustion chamber already containing the burning gases at high temperature and at an advanced stage of combustion. When this burning of the remaining fuel occurs, the flame spreads throughout the combustion chamber and the ensuing increase in pressure and temperature is so great that the rate of combustion is accelerated sufficiently to burn the fuel with practically *no ignition delay* as it comes from the injection nozzle. A further rise in pressure then occurs, reaching a peak, or the conditions are such as to maintain a *constant pressure* during the remainder of the injection and combustion period. It is evident that this last period or phase of burning is under direct control of the mechanical operation of the fuel system. These three phases or stages, as shown in Fig. 5-17, are marked off on the curve, the dotted line corresponding to the expansion line if no fuel were injected. According to the law of perfect gases, assuming no heat losses to the cylinder walls, this curve would be a looking-glass reproduction of the compression curve, or what is known as *adiabatic expansion.*

Elements of fuel injection. The fuel pump and spray nozzle must perform the important functions involved in the process

of injection. These units replace the carburetor and the spark plug in the gasoline engine. It can be seen that the operation of the fuel pump during injection of the fuel takes place within a few thousandths of a second. During this brief interval of time the pump must build up a pressure of several thousand pounds per square inch, discharge the fuel over several degrees of crank angle, return to a state of rest, and be ready to make the next stroke. Under a pressure which may vary from 1500 to 10,000 psi, accurately controlled and measured quantities of fuel, so small in some engines that practically no leakage is permissible, must be delivered in such a short interval of time that a few degrees of variation at the beginning of injection and cutoff will affect seriously the operation of the engine and its economy.

In order to visualize this operation of the fuel pump as to pressures, time of injection, and its relation to combustion efficiency, examine Fig. 5-18. As shown, the pressure rises from

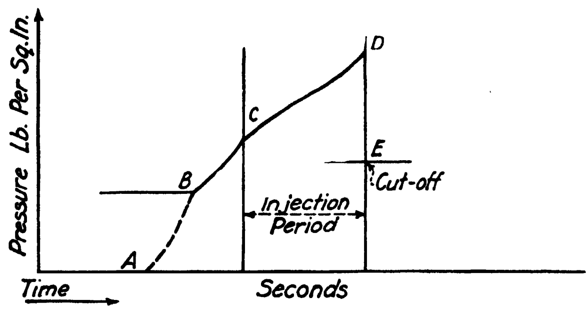


Fig. 5-18. Pressure-time diagram of fuel injection.

zero to the pressure required for proper atomization and penetration of the fuel charge at a very rapid rate and falls back or cuts off quickly enough to prevent drip of fuel after injection is finished. It is evident that these mechanical requirements imposed upon the injection pump require that this mechanical device be accurately constructed according to high standards of precision. The fuel injection pump is a volume-changing device, designed to perform two important functions of measuring the fuel and delivering it to the injection nozzle under high pressure sufficient to force it through the spray valve with sufficient velocity to atomize the fuel. It is obvious that these two functions of the fuel injection pump are possible only if the

pump has the following characteristics of design and operation:

1. The fuel injection unit, comprising the pump and spray nozzle, must be able to introduce fuel into the combustion chamber in a finely divided or atomized state, suitable for rapid burning and easy ignition.

2. It must be able to spread the atomized fuel into the chamber in such a manner that it will mix intimately with the air for combustion.

A careful distinction should be made here between a thoroughly vaporized mixture and an atomized mixture. The fuel mixture of the gasoline engine is a complete vapor, the gasoline having changed from a liquid to a gas. This is not the case with the Diesel fuel charge. The Diesel fuel is atomized through the spray nozzle into finely divided droplets, or globules, but it is still in a liquid state. These droplets begin to vaporize when heated, and ignition commences from the *surface of the globules.* The Diesel engine injection system does not convert the Diesel fuel charge into a perfect vapor but depends upon various conditions in the combustion chamber to accomplish this purpose. Even with gas fuel in a Diesel engine, a *pilot quantity of fuel oil* is injected with the charge of gas to furnish the initial ignition and insure its reliability.

The fuel injection unit must be capable of controlling both the timing and the rate of admission of the fuel so that no undesirable peak pressures will be produced during the combustion period.

The two principal methods of fuel injection now used on the Diesel engine are known as the *jerk-pump* system and the *common-rail* system, already referred to. The jerk-pump system introduces the fuel through spring-loaded automatic injection valves or nozzles by means of an individual pump for each cylinder. This system is sometimes called the *timed pump* as opposed to the time valve of the common-rail system.

Since the jerk-pump system is more generally used, this system will be described more in detail. The timing is effected by the pump, which delivers the fuel to the injection nozzle only during the injection period. In Fig. 5-19 is shown the essential features of design of the modern jerk pump. It is operated by an engine driven cam, the setting of which controls the timing and the metering, or measuring, of the fuel. The differential needle atomizer, or injection valve as it is commonly

called, shown at the right in Fig. 5-19, definitely controls the beginning and the end of injection by means of its spring-loaded needle. It is hydraulically operated, that is, by means of the pressure built up by the pump. The parts of the pump are indicated. As shown in the pressure-time diagram, Fig. 5-18, the ideal pressure requirements for the jerk-pump system are low initial rates of injection with increasing rates until the maximum is reached at the time injection ceases at cutoff.

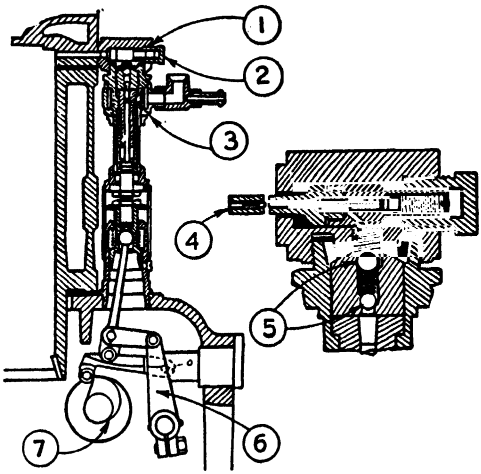


FIG. 5-19. The Dorner fuel pump shown at the left is a variable-stroke pump with constant-throw cam. The enlarged construction drawing at the right shows details of the injection valve. (1) Injection valve; (2) enlarged section of injection valve; (3) pump; (4) nozzle; (5) ballchecks; (6) pump control linkage; (7) cam.

The ideal *pressure-time diagram* is a close approximation of that obtained by the indicator. The pressure in the pump working barrel begins to rise upon the seating of the suction valve or the closing of the suction port at A, Fig. 5-18. At B, the pressure in the pump chamber has risen to equal that of the fuel remaining in the delivery line from the last cycle of the pump, the delivery valve then lifting off its seat. This pressure then begins to rise further until the atomizer valve lifts at C, where injection begins. The pressure then continues to rise to point D, when the cutoff valve operates to relieve the pressure that declines to a value lower than the closing pressure of the springs in the injection spray nozzle, thus giving a sharp cutoff of the injection.

There are many variations of the jerk-pump system; the principal classifications are:

1. Part of Plunger Stroke Used
   A. Start fixed end of stroke by-passed
      (1) Valve type
      (2) Port type

B. Start and end of stroke by-passed
(1) Valve type
(2) Port type
2. Whole or plunger stroke used:
A. Variable rate of plunger life
(1) Guiberson
(2) Dorner
B. Variable plunger clearance
(1) Numerous variations
3. Spring injection

An advantage of the jerk-pump system is that the method of distribution renders it less difficult to supply the correct amount or volume of fuel to each cylinder although the speed of the pump is greater than when the common-rail system is employed.

Classes of fuel pumps. The jerk pump that utilizes the entire stroke of the plunger or the fuel pump is open to certain objections that do not exist when only a portion of the stroke of the plunger is used. In this first instance, the plunger must start and stop at the beginning and the end, respectively, of each of its strokes; the initial and final deliveries must be relatively slow. The slow final delivery usually is a drawback on the high-speed engines, for it is most essential to have quick cutoff in order to produce a rapid combustion with low peak pressures; the nozzle should not dribble, as it does with slow cutoff.

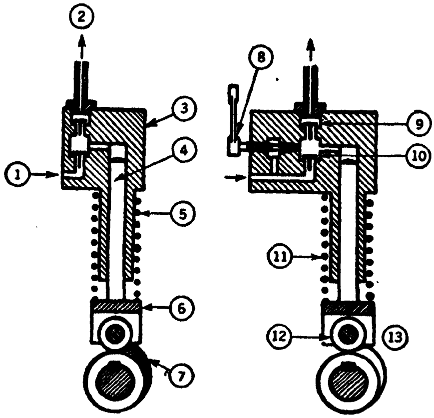


FIG. 5-20. Two methods of regulating fuel delivery from a pump. That at the left has a variable profile cam that varies the plunger stroke. The pump shown at the right is provided with a needle valve that permits regulation of the by-pass and therefore provides for changing the net delivery of the pump. (1) Fuel inlet; (2) fuel to injector; (3) pump body; (4) plunger; (5) pump cylinder spring; (6) cam follower; (7) variable profile cam; (8) needle valve control; (9) outlet check; (10) inlet check; (11) plunger return spring; (12) cam roller; (13) cam.

A well-known fuel pump is provided with a variable amount of lift by altering mechanically the clearance between the plunger and its operating cam. In this instance the suction and discharge delivery valves are both automatic, and this type of pump is known as the *variable-stroke pump*. There is a disadvantage, however, to this method in that the load is decreased by reducing the amount of lift of the plunger, which retards the timing of the injection to a certain extent unless the pump is designed to prevent it. It is desirable actually to advance the timing under light loads. However, in one fuel pump of well-known design (*Guiberson*), an automatic control to advance or retard the cam on its shaft for light and full-load conditions has been developed. When only part of the stroke of the pump is used for injection, the disadvantage of the full-stroke method is avoided. In other instances, pumps cut off at the stop end only. Such a type of cutoff is suitable for medium-speed engines. For high-speed work, it is desirable that both ends of the stroke be cut off, the central portion only being used to give the injection a high velocity throughout the injection period. In this instance, the plunger has a much better uniform motion over as much as possible of its stroke. The start of the injection is controlled by closing the pump port; the end of the injection is controlled by opening a control port in the pump barrel. With this type of control, a piston valve is used that overruns a port cut in the pump cylinder, or guide. The piston valve may be either the plunger itself, or a special piston valve. The port is usually in the form of a round hole drilled in the guide. When the pump valves are thus controlled, they usually are operated from the camshaft by means

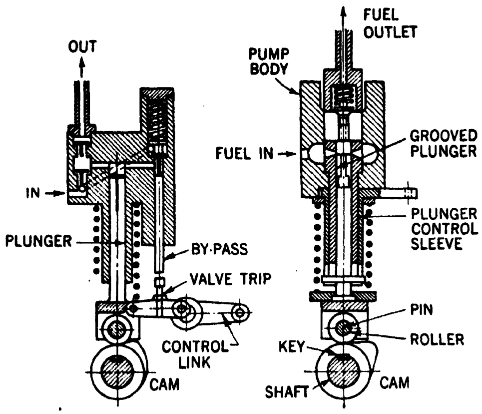


FIG. 5-21. Other methods of regulating fuel pump delivery. These pumps have a by-pass regulation. That shown at the left has a valve that can be tripped at any desired point in the plunger stroke. The design at the right has a sloping groove in the plunger.

of rocker arms or a rocking lever. An essential feature of the mechanism is that it must be kept absolutely rigid; otherwise there will be a variation in the timing and metering of the fuel to the nozzle.

It is therefore evident that the fundamental factors in the design of the pump are the means employed for controlling the quantity of the fuel and the timing of its injection. There are two methods of doing this, as shown in Fig. 5-20. In one of these, two different means are employed:

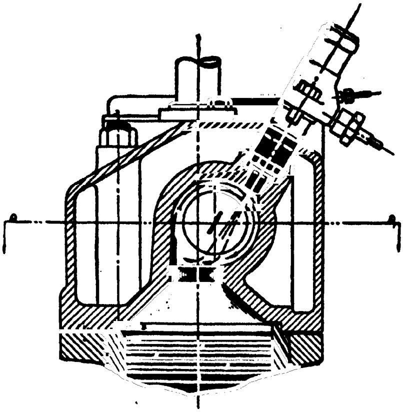


FIG. 5-22. The Palmer Marine Diesel, vertical cross section through a patented cylinder head illustrating the tangential injection of the fuel into the combustion chamber. The exhaust and air intake valves are arranged horizontally.

1. A sliding cam that varies the lift of the plunger, thus shortening its stroke. The cam has a suitably varied outline that may be set to any degree of variation that suits the designer's purpose.

2. Needle valve regulation. Part of the fuel entrapped in the pump during the working stroke of the plunger is permitted to escape through a by-pass to the fuel intake of the pump. The quantity by-passed and therefore the amount of fuel permitted to reach the nozzle, is regulated by means of a throttling pin or needle.

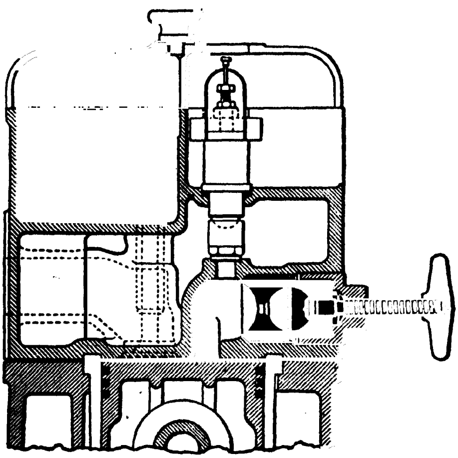


FIG. 5-23. Section through the cylinder head, showing the combustion chamber with air cell and valve used when starting.

The other method is to have a by-pass port open after the plunger has traveled a definite distance, the port being in connection with the

fuel intake line. In practice this is accomplished by opening a special by-pass valve, or the inlet valve of the pump itself, for a longer or shorter period before the end of the working stroke of the plunger, as shown in Fig. 5-21. Pumps without suction valves, in which the piston or plunger itself controls the intake port, have a slanting groove in either the piston or the piston sleeve, and this groove registers with the opening in the piston sleeve or the piston earlier or later in the

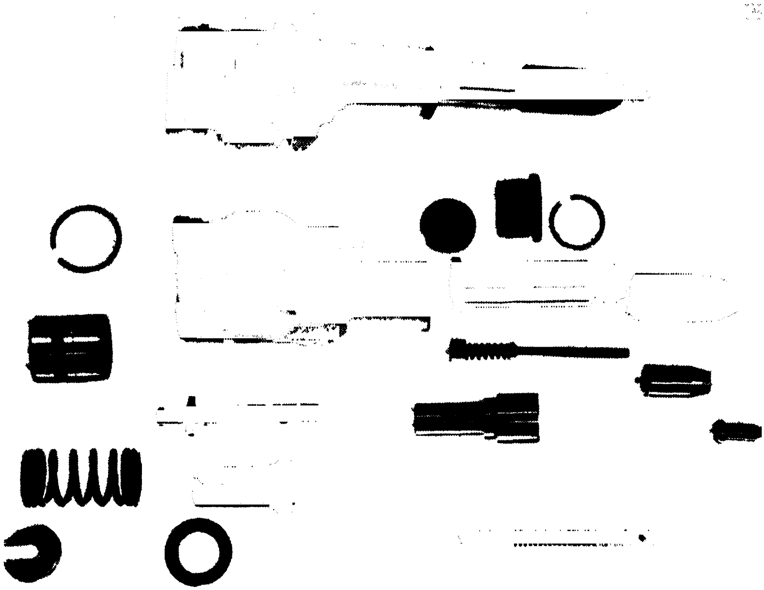


FIG. 5-24. Exploded view of fuel pump and injector for Hendy (Series 20) engine.

working stroke, according to the relative load. Thus it can be seen that with an engine operating at 2000 rpm, the injection period being but 20 to 30 deg of the crank takes place in approximately 0.002 sec.

The quantity of fuel injected per stroke of the fuel pump for a 50 hp engine is of the order of 0.004 to 0.05 cu in. at normal loads, and 0.001 to 0.002 cu in. at light loads. Fuel is injected through the spray nozzle for the purpose of atomizing it, the nozzle having orifices varying from 0.008 to 0.045 in. in diameter. The length of these holes is approximately 0.02 to 0.04 in. The injection pressures may vary over a wide range, with some

as high as 10,000 psi, but usually for practical purposes in commercial engines from 1200 to 3000 psi.

It is obvious therefore that the fuel injection unit of the Diesel engine deserves careful study and complete understanding by the operating man. Understanding these elements of

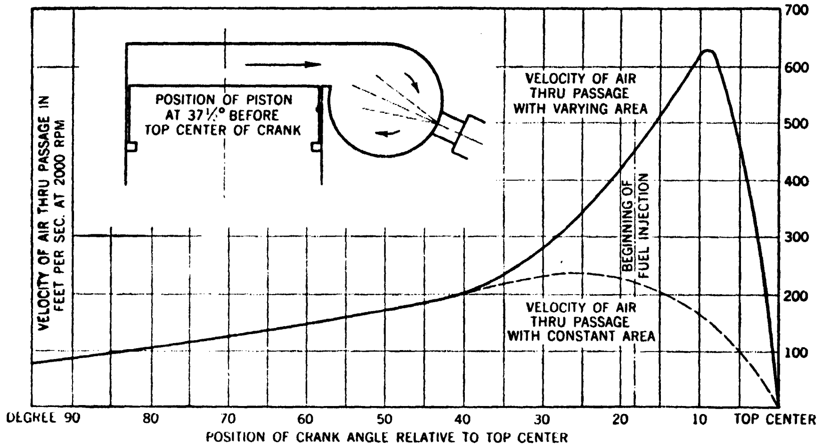


FIG. 5-25. Chart illustrating air velocity in the combustion chamber with respect to piston position.

the fuel injection system and what they are designed to accomplish is the first step toward learning the operation of the Diesel engine.

QUESTIONS

1. What is the function of turbulence during the compression stroke?

2. In what designs is turbulence created before the fuel is injected?

3. What features do all combustion systems have in common?

4. What system provides the highest mean effective pressure?

5. What feature of the precombustion chamber restricts the efficiency of heat utilization and mean effective pressures?

6. What effect do the inert gases trapped in the precombustion chamber have on volumetric efficiency?

7. What are the advantages of the precombustion chamber?

8. What are the three major groups of chambers?

9. Are multiple-orifice injection nozzles used with open or single combustion chambers, or with precombustion chambers?

10. What is controlled turbulence?

11. Did Ricardo's design employ a multiple-orifice, or a single fuel jet for the injection of the fuel?

12. What are the disadvantages and advantages respectively of the multiple-orifice and single-hole orifice injection nozzles?

13. In what class of engines is good idling essential and what types of combustion chambers are usually used in such engines?

14. In what type of combustion chamber is there the least heat loss?

15. What type of combustion chamber may be started with lower compression pressures?

16. What are the two positions for locating the fuel injection nozzles?

17. Why is the heating of the fuel objectionable in the gasoline engine and desirable in the Diesel engine?

18. Why does the amount of air pumped limit the output of the Diesel?

19. What two problems are involved in the fuel-mixing process?

20. A delay of the combustion process produces after-burning in the Diesel engine? What causes the delay?

21. What is the result of after-burning?

22. Upon what does the time lag of ignition and the delay of combustion depend?

23. Does a partial ignition occur in some precombustion chambers? In what type?

24. How does the Lanova system of combustion "control" the turbulence?

25. Why is the method of rotating the combustion air to obtain complete combustion called "organized combustion" in the Ricardo Comet type of chamber?

26. Why does the fuel start burning from the "surface of the droplets"?

27. Who named the so-called "three stages of combustion"?

28. In what stage does ignition delay occur?

29. Is the delay period constant in terms of time or degrees of crank angle?

30. By what means have the designers been able to shorten the period of ignition lag?

31. Is the rate of pressure rise during the second phase of combustion constant in terms of crank angle or time as is ignition lag in the first stage of combustion?

32. Does "delay period" mean the same as "ignition lag"?

33. What is the difference between a vaporized mixture and an atomized mixture?

34. When does the fuel in the Diesel engine commence to vaporize, and how does vaporization start?

35. Does the Diesel convert the spray into a perfect vapor, or just an atomized spray consisting of minute droplets or globules?

36. Name the classes of timed-pump injection systems.

37. What are the two means of controlling the quantity of fuel injected, and the timing of the injection?

38. In the variable-stroke fuel pump, are the suction and discharge valves both automatic?

39. What is the advantage of a quick cutoff of the fuel pump?

40. What objections to the use of the entire stroke of the plunger in constant-stroke pumps do not apply when only a portion of the stroke of the plunger is used?

41. Which combustion-chamber types would be preferred for: (a) Power output? (b) Fuel economy? (c) Ease of starting? (d) Speed flexibility? (e) Smoothness of combustion?

42. The divided combustion chamber is harder to start because of what?

43. What happens if the fuel is burned with no excess air in a gasoline engine? In a Diesel engine?

44. What are three effective methods of increasing the output of a given displacement Diesel engine?

45. What type of combustion chamber is most efficiently scavenged?

46. How does the preignition chamber help to improve the combustion of the fuel? Does it have a higher temperature than the main combustion chamber walls?

47. What effect can the preignition chamber have on the time lag of ignition? Does it create a pilot quantity of fuel vaporization that will reduce the time lag of ignition?

48. What is the function of the larger antechamber? Does it serve as a vaporizer and ignition chamber in the same way that a smaller preignition chamber or air-cell chamber does?

49. What is the function and principle of the small antechamber? Does it contain sufficient air to burn the fuel injected into it or only part of it?

50. When the neck of the chamber is made small, how does this aid in mixing the fuel? Does the air trapped in the chamber during compression receive fuel and initiate the ignition?

51. What are the general functions of the antechamber types?

52. Are antechambers, or precombustion chambers, better suited to giving flexibility? Why?

53. Is there more or less heat loss in the antechamber than in the open combustion chamber?

54. In what design of combustion chamber is better or higher mean effective pressure realized? Why?

55. Is the antechamber as well scavenged as the open chamber?

56. What design affords easier starting when cold?

57. Is the surface to volume ratio greater for the antechamber than for the open combustion chamber?

58. Which class of chambers is better suited to the burning of heavier fuels? Which design will more nearly burn up the residues?

59. Which type of chamber more nearly takes care of the heat changes due to variations in mean effective pressures? Is the best heat regulation related to better scavenging of the open combustion chamber?

60. Can the path and characteristics of turbulence from a preignition chamber be varied as much and as easily as with single combustion chambers of the open design?

61. What chamber design is better suited for the regulated variation of the fuel discharge during injection?

62. What is the chief disadvantage to the two-stage combustion process?

63. Does the reduction of engine size limit permissible design features that have been a benefit on larger engines?

64. What chamber types give higher cyclic efficiency? Greater ratio of expansion? Higher cylinder output and efficiency?

65. What functions are performed by the precombustion chamber and main chamber that cannot be performed by the single, open combustion chamber?

66. What supplementary features may be necessary in order to obtain flexibility from the single combustion chambers?

67. What are the three states of combustion? Who was Ricardo?

68. Why is it desirable to have the fuel spray uniform?

69. What effect has the fuel injection pressure on the size of the droplets?

70. What other factors determine the droplet size?

CHAPTER 6

BASIC MAINTENANCE PROBLEMS

Introduction. Procedures for teardown inspection, repair and cleaning, and reassembly of the engine apply to both stationary and marine engines. A record of all work done, including a log of all measurements and adjustments, should be written. The purpose is to aid the operator and maintenance man in working out his own procedure for the upkeep and overhaul of his own engine.

It matters little how simple or complicated the program may be, the responsibility for repairs must be vested in some person or persons who are able and willing to undertake it.

Attitude toward maintenance. There seems to be two schools of thought relative to maintenance of engines and other machinery. One group advocates the repair and overhaul of engines if and when they will no longer run. The engines are allowed to run without a program of systematic inspection and rehabilitation. This view is known as the "fix it when it quits" practice. This practice has largely been supplanted by systematic and well-planned programs, sometimes referred to as "preventive maintenance." What it really prevents is a disaster to the engine when things are neglected too long. It's a stitch in time.

The objections to a "hit or miss" program for maintenance are obvious. Regardless of how effective and elaborate the other features of the program, it fails unless the actual work is carried out correctly and systematically. General rehabilitation of the engine parts periodically is well-established practice. Experience has shown that periodic inspection, adjustment, and repairs are the factors that contribute to economy, satisfactory operation, and dependable service.

Period between overhauls. When operating conditions are fairly constant, the most economical operating period between engine overhauls can be determined. The influence of operating conditions on overhaul time has been indicated. The following major factors are considered when determining the period of operation between major overhauls:

1. Load factor and frequency of starting and stopping.
2. Lubrication, fuel used, and climatic conditions.
3. Practice in making current repairs and minor adjustments.
4. The application and type of engine.

The study of adequate records of operating conditions will indicate the proper periods between major overhauls. In some cases, it has been found that a period of 8000 to 10,000 hr of operation for heavy-duty stationary engines in well-administrated power plants produces satisfactory results. A similar consideration of all available operating information will provide the basis for systematizing inspection periods for the major engines of a marine installation. This procedure of setting up a definite program for maintenance should be extended to other more frequent maintenance work on parts that need maintenance between major overhauls. Fuel pumps will produce better results if serviced at more frequent intervals. However, each individual maintenance operation should be put on a schedule, and a list for each set of conditions should be set up and followed as closely as judgment dictates.

Standard procedure. A standard procedure is one of continued growth, improvision, and development. The last word on the subject cannot be given here. It would be an infinite task to detail the method to be followed on each repair job or inspection. However, standards are set up for major overhauls. A complete list of the inspections made during a period of major overhaul is included in Chapter 7 but not repair detail. Certain forms for reports of inspection and checking are included.

The operator and maintenance engineer making the actual inspection should be provided with an outline covering each particular job. These outlines should be supplemented with specific instructions on the repair methods dealing with the more complicated jobs, such as checking crankshafts and other alignments, liner wear, clearance, valve adjustment, etc. An important contribution to this field of engine maintenance was

made by E. R. Spencer, Engineering Department, The Cooper-Bessêmer Corporation, and published by the *Petroleum Engineer*, to which the author is indebted for the outline of these topics. Some of these topics are discussed in Chapter 7. If such an outline is made for various types of equipment in a power plant, in the light of existing data and tempered by qualified experience, Spencer points out, its value is great, especially where a large number of engines are operating under one maintenance engineer or supervisor. It gives the supervisor of maintenance a definite standard to direct those who actually make the inspection and have the responsibility for the maintenance and repair work. The wide variation of judgment among mechanics makes the use of such instructions a practical necessity if the maintenance work and program is to be carried out systematically and effectively. The assurance of complete and thorough inspection of the equipment will more than justify the work and effort involved. This and the following chapters will indicate the use of such a standardized program to be followed in making inspections and repairs. Not all the details of each operation can be given here. In subsequent chapters, additional discussion of the details to cover major parts is presented for the purpose of assisting the maintenance man in adapting it to his own problems.

Any supplementary information relative to the repair and maintenance of any particular unit of the equipment should be made available to the men having charge of the operation and maintenance. Standard adjustment information is particularly needed. The following list give some information relative to the wide range of topics that should be tabulated for information on any application: (1) back lash in gears, (2) side clearance in piston rings, (3) piston ring gap, (4) piston head clearance, (5) valve lift, (6) valve dimensions, (7) maximum and minimum temperatures, (8) compression and firing pressures, (9) operating speeds, (10) bearing clearance, (11) standard over and under sizes, and so on.

A table of such information is given herewith (Table 6-1), showing the new and worn clearances, or wear limit for one engine. When examining this table and taking into account the larger number of different types of information needed, the considerable amount of data required to make the maintenance work effective is evident; yet the operator and the maintenance man must have this information, and must know how to use it.

Major top overhaul procedure. This includes a step-by-step operation for proper inspection:

1. A routine inspection of the intake and exhaust valves must be carried out. The kind of fuel used and the load on the engine will influence the need for this inspection.

2. Check the rocker arm bore for wear by inserting the fulcrum pin and then measuring the exact amount of wear by means of a feeler gauge. Excessive wear in fulcrum parts should be corrected either by new parts or by remachining the

TABLE 6-1

DIMENSIONS, CLEARANCES, AND WEAR LIMITS
COOPER-BESSEMER MARINE DIESEL ENGINE

MAIN BEARINGS	
Shell to shaft clearance (new)	.006–.008 in.
Shell to shaft clearance (max. allowable worn)	.012 in.
Minimum allowable shell thickness (worn)	.493 in.
Thrust bearing end clearance (new)	.005–.007 in.
Thrust bearing end clearance (max. allowable worn)	.018 in.
Wedge screw torque	125# ft.
No. 10 main brg. torque, nut, lower half	350# ft.
No. 10 main brg. torque, nut, upper half	120–175# ft.
CONNECTING ROD	
Shell to shaft clearance (new)	.006–.008 in.
Shell to shaft clearance (max. allowable worn)	.012 in.
Minimum allowable shell thickness (worn)	.368 in.
Crankpin diameter (new)	7.498–7.500 in.
Piston pin diameter (new)	3.9965–3.9975 in.
Minimum allowable piston pin diameter (worn)	3.992 in.
Connecting rod bolt torque	140# ft.
PISTON AND LINER	
Piston to liner clearance (new)	
(a) At top of piston	.063–.068 in.
(b) At skirt	.016–.019 in.
Maximum allowable piston to liner clearance (worn, at skirt of piston)	.030 in.
Piston measurements (new)	
1. At crown	10.434–10.437 in.
(a) Between No. 1 and 3 ring grooves	10.455–10.457 in.
(b) Between No. 3 and 5 ring grooves	10.464–10.466 in.
(c) Between No. 5 and 2 oil rings	10.473–10.475 in.
(d) Piston skirt	10.483–10.484 in.
Minimum allowable piston diameter (worn) junction of taper, below fifth ring groove	10.468 in.
Cylinder liner bore (new)	10.500–10.502 in.
Maximum allowable liner diameter (worn)	10.550 in.
Maximum allowable liner out-of-roundness (worn)	.010 in.
Piston pin bore (new)	3.9995–4.0005 in.
Maxium allowable piston pin bore diameter (worn)	4.005 in.
Piston pin to piston pin bore clearance (new)	.002–.004 in.
Maximum allowable piston pin to piston pin bore clearance (worn)	.010 in.

TABLE 6-1 (*Cont.*)

PISTON RINGS	
Compression gap clearance (new)	.045 in.
Compression gap clearance (max. allowable worn)	.125 in.
Oil control gap clearance (new)	.025 in.
Oil control gap clearance (max. allowable worn)	.125 in.
Compression side clearance (new)	
(a) No. 1 and 2	.009–.012 in.
(b) No. 3, 4, and 5	.004–.007 in.
Compression side clearance (max. allowable worn)	
(a) No. 1 and 2	.015 in.
(b) No. 3, 4, and 5	.010 in.
Oil ring side clearance (new)	.004–.007 in.
Oil ring side clearance (max. allowable worn)	.010 in.
CYLINDER HEAD	
Relief valve setting (hydrostatic pressure)	2250 psi
Cylinder head to block nut torque	1000# ft.
Cylinder head to liner cap nut torque	95–100# ft.
CAMSHAFT	
Minimum allowable bearing shell thickness (worn)	.064 in.
Bearing shell to shaft clearance (new)	.002–.004 in.
Maximum allowable end clearance (worn)	.012 in.
End clearance (new)	002 in.
Maximum allowable bearing shell to shaft clearance (worn)	012 in.
VALVES	
Exhaust valve tappet clearance (cold)	.020 in.
Inlet valve tappet clearance (cold)	.020 in.
Inlet valve guide to valve stem clearance (new)	.003–.005 in.
Maximum allowable inlet valve guide to valve stem clearance (worn)	.010 in.
Exhaust valve guide to valve stem clearance (new)	.006–.008 in.
Maximum allowable exhaust valve guide to valve stem clearance (worn)	.015 in.
Inlet valve stem diameter	.746–.747 in.
Exhaust valve stem diameter (new)	.743–.744 in.
MISCELLANEOUS	
Injection nozzle opening pressure	3000 psi
Lube oil recommended	SAE 40
Lube oil recommended (first choice substitute)	SAE 30

fulcrum and rebushing the rocker arm. Inspect the cam rollers for flat surfaces, and check the roller pins with micrometers for out-of-round. Accurate measurement of wear between roller and pin should be obtained with a dial indicator attached to the rocker arm with the dial pin resting on the roller. Repair or replacement of these parts is justified by the wear limit requirements.

3. Tappet screws and tappet bearings, contact surfaces, and plates should be checked for wear and *fracture* of case-hardened surfaces. When the parts are worn sufficiently to interfere with valve action, the bearing plate should be replaced and the screw repaired or replaced.

4. After the valves are removed from the engine, the seating surfaces should be inspected to determine whether *leakage* of gas is evident. Leakage may be the result of a warped valve, dirty valve seat, a worn seat, or even a bad gasket. Determine the cause of any valve leak.

5. The seating surface in the cylinder head should be checked for *warping*. This can be accomplished by "bluing" and turning a new seat on its corresponding seat in the cylinder head. If this surface is warped, it may be ground in with the cylinder head in place by using a machined plate of the same diameter as the seat. In some cases the seat may be remachined on the job by a special portable seat-cutting machine.

6. Any deposit of *carbon* on the valve stem should be noted as this may indicate a worn guide bushing. After the valves have been dismantled, the valve stem *bushing wear* can be accurately measured, and if the repair is necessary or the wear limit exceeded, the bushing should be replaced with a new one, that is, the cage bored out and a new bushing installed. Also inspect the valve stems.

7. If the valve seats and discs are *pitted* badly or worn enough to form *shoulders*, they should be refaced before attempting to grind them. When the valves have been cleaned, the seat and cage should be carefully examined for warped surfaces. A set of thickness gauges may be used to detect warping or the seat may be "blued" and the valve rotated on the seat in position. If there is any doubt as to the condition of the seat, a new seat or one that has been refaced should be used. When there are indications of warped surfaces found, all parts must be refaced or remachined.

8. Before a valve is ground to its seat in the cage, the valve cage should be placed in a vertical position, with the seat on top of a table or bench. The valve is then inserted and the two seating surfaces are ground together. A medium grade of grinding compound should be used until the seats are free of pits and burrs; then use a fine grinding compound to polish the two surfaces. After the valve is ground and polished, it should again be "blued" and rotated on its seat to test for complete surface contact between the seat and the valve.

9. The valve mechanism can now be assembled, special attention being given to the spring length and tension. It is evident that the removal of the metal from the valve seat and cage during the grinding and polishing and truing up or machin-

ing process materially affects the tension of the spring when it is in place. Washers placed under the springs to give a thickness equal to the metal removed will keep the spring tension normal.

10. The valve assembly is now ready to be placed back in the cylinder head. The seat in the head should be cleaned and inspected for cracks. If gaskets are used, place a new gasket on the valve cage seat. Care should be used when placing the valve in the cylinder head, inspecting the seat in the head, and cleaning the surfaces. Use care to avoid damage to the gasket. The proper installation of the valve cage in the cylinder is very important.

11. The valve assembly is now in the head. The valve cage nuts should be tightened just enough to prevent gas leakage. Any excessive strain placed on the cage-type valve by over or uneven tightening of the cage nuts will distort the valve assembly, thus preventing it from functioning properly. When cylinder heads have been removed and time permits, the final grinding should be completed, after the cage has been installed in the cylinder head. This procedure eliminates the possibility of seat-warpage caused by tightening the cage studs.

12. After valves and rocker arms are installed in the cylinder, the correct rocker arm-to-roller cam clearance should be set as recommended by the engine manufacturers.

Checking alignment and bearing clearances. In all engines it is very important that the crankpin and piston pin bearings be true in every respect. The bore of each bearing must be parallel to the crankpin in order to obtain satisfactory operation. In order to do this careful technique must be learned and practiced.

Methods of checking alignment. There are several methods for checking the bore and alignment of these bearings. The most common method of checking the crankpin bearing is shown in Fig. 6-1. The mandril or "dummy" pin is machined to the exact diameter of the crankpin and long enough to allow suitable overhang at each end for measuring purposes. The bearing is then placed on a surface plate supported by two identical and accurately machined metal strips in order to clear the "boss" that is present on many bearings of this kind. The mandril is then placed in the bearing, and by means of a surface gauge with a dial indicator attached, one can set the dial indicator and move it directly across the mandril to observe the maximum dial readings. This operation with the same dial

setting is repeated on the other end of the mandril and the maximum readings again observed. The two readings should

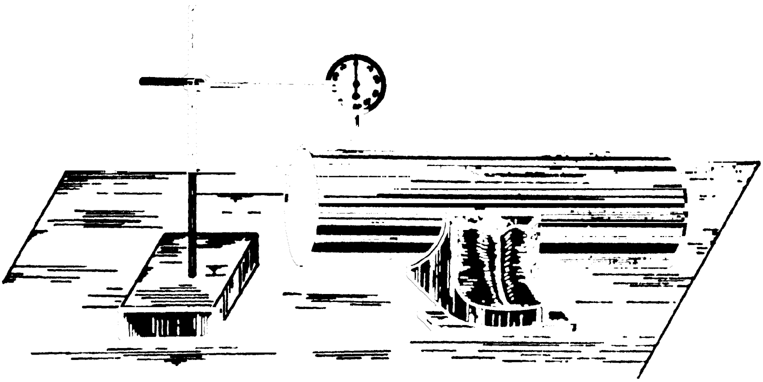


FIG. 6-1. The most common method of checking the crankpin bearing.

be the same. An ordinary surface pin gauge will serve the purpose of checking the parallelism of the mandril and surface plate. When dial indicators or micrometer calipers are used, however, one can determine the amount of taper in the bearing, should the bore not be true.

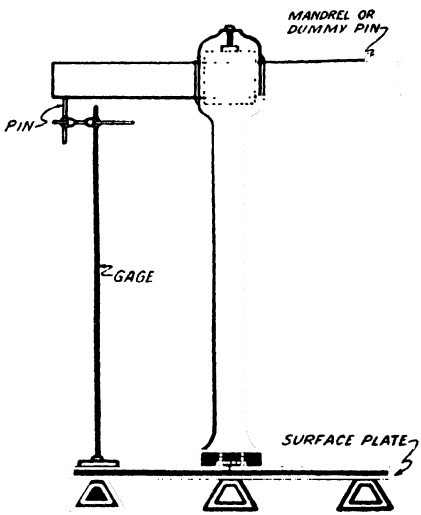


FIG. 6-2. One method of checking the bore of the piston-pin bearing relative to the large end of the connecting rod.

It is good practice all through the operation to "blue" the mandril and rotate it in the bearing to check the seating surface before the above tests are made. There may be burrs or dirt and carbon that will need scraping away in order to allow the mandril to rest in the entire seat of the bearing. This will eliminate the possibility of erroneous readings.

It is important that the piston pin bore as well as the crank bearing bore be parallel to the crankpin. In Fig. 6-2 is shown the method of checking the bore of the piston pin bearing

relative to the big end of the connecting rod. A mandril is machined to the exact diameter of the piston pin. The mandril is "blued" and the seating surface of the bearing checked; then the mandril is clamped in place by the adjusting screw; the bearing halves, less the shims and the rod, are placed in a standing position on the surface plate. By means of a gauge that consists of a rod of sufficient length to accommodate small micrometer calipers or pin assembly at one end and a surface block at the other, the distance can be determined between the surface plate and each end of the mandril or the "dummy" pin. This will readily show the alignment of the bearing bore

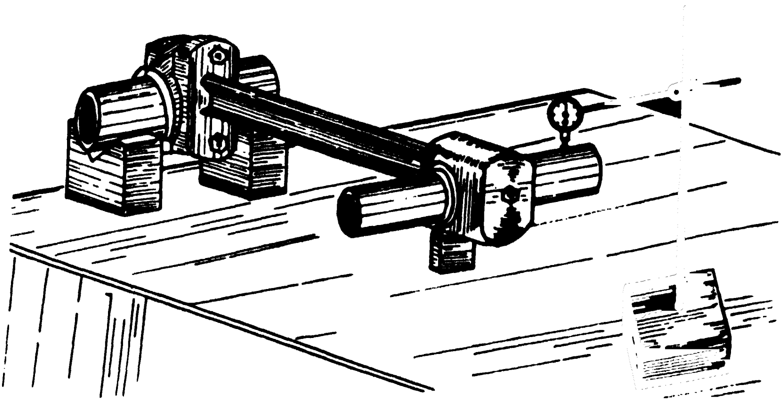


FIG. 6-3. A method commonly used to check the parallelism of the bearing bores relative to each other.

with respect to the bottom of the connecting rod. An ordinary pin gauge will suffice for this check when micrometers are not available. The advantage of using micrometers is in the quick adjustment and in ascertaining the exact amount of taper in the event the bore is not true with the bottom of the piston rod.

The bore of the piston pin and crankpin bearings must be in a horizontal position parallel to the crankpin, yet the two bearing bores must not be in the same vertical plane. This condition may occur when the connecting rod is twisted or when either or both the piston pin and crankpin bearings are not bored true. Engines operated in this condition give trouble and this should not be tolerated, for it is detrimental to smooth operation.

Fig. 6-3 shows a method of checking the parallelism of bearing bores relative to each other. Both mandrils are clamped

in the bearings with the bearing halves or clamps, but without shims; the rod is then laid on the surface plate with two V blocks supporting the crankpin mandril. The two V blocks must be accurate in construction and identical in order to get results. A check on the V blocks can be obtained by measuring from the surface place to the mandril on the top at each end with the same gauge as used in checking the crankpin bearing in Fig. 6-1. If the distances between the surface plate and the top of the mandril of each end of the piston pin mandril are equal, the two mandrils are in the same plane.

Much time could be spent in discussing the bearing and rod alignment because a different method of checking is applicable to each of many types of engines and installations. The method here described is largely applicable to heavy-duty, slow-speed, vertical engines that are found in many stationary and marine applications. For this reason the method is described in detail and illustrated as an example of procedure. It is also evidence of the systematic maintenance methods in use.

Alignment of piston. A great deal of trouble is encountered in the engine if the piston does not align with its adjacent parts, the piston pin bore, rods, and bearings. In Fig. 6-4 is illustrated a method of checking the alignment of piston and rod with the connecting rod in place. This is easy enough on a stationary engine, sitting level. The piston is placed on its head and leveled; the rod is placed in its exact parallel-vertical position; and by means of a straight edge alongside the piston skirt perpendicular to the rod movement, the distances X and Y are noted.

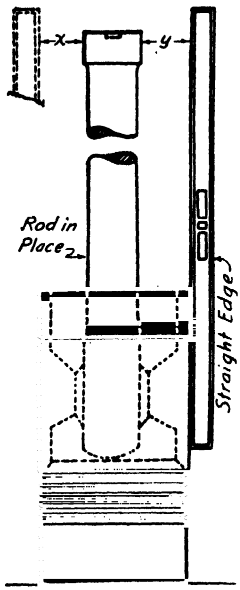


FIG. 6-4. A method of checking alignment of a piston and rod with a connecting rod in place.

These two dimensions must be equal, provided the side clearance in the piston pin bearing is divided equally. To determine the exact location of the straight edge, in order to measure the X and Y perpendicular to the rod movement, scribe a line on the piston skirt with a try square vertically through the center of the piston pin on both sides of the piston. A straight edge directly over these lines will locate it in the

center of the piston pin and 90° with the rod movement. This method will also give a check on the piston pin bore or a possible tapered pin, provided the pin bearing has been checked for true bore. A tapered condition is usually determined by measuring with a micrometer.

Another procedure for checking the piston bore through the piston is shown in Fig. 6-5. The piston, with the pin in place, is leveled on its head by a level and a combination tool or by a straight edge and a level across the end of the piston skirt. Wedges under the piston head will simplify the leveling of the piston. A level "L" on the pin will reveal any variation in pin alignment, relative to the piston, which may be attributed to an untrue pin bore.

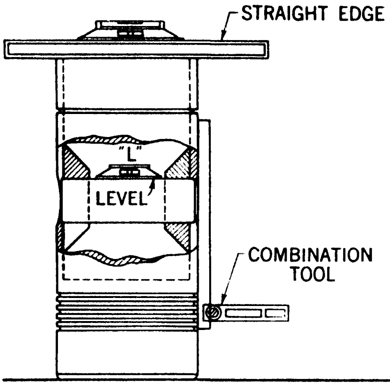


FIG. 6-5. Another way of checking piston bore through piston.

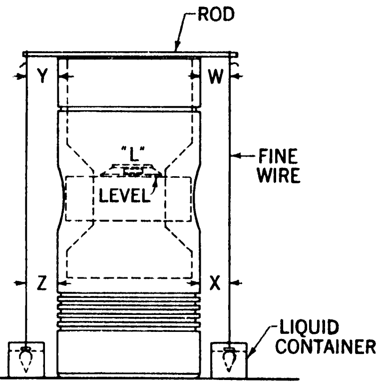


FIG. 6-6. Method of leveling a piston.

Fig. 6-6 shows how a piston can be leveled by suspending heavy weights or plumb bobs by a fine rod from the rod placed across the end of the skirt parallel and then perpendicular to the axis of the pin. Wedges are moved under the piston until the dimensions *W* and *Y* measured by micrometers or pin gauge are equal. Dimensions *Y* and *A* then should be equal and represent a check of other readings. The color "*L*" should show the pin as being perfectly level if no defects are found. The weight of plumb bobs have a tendency to swing when subject to wind or to light handling. This tendency to swing can be overcome to a great extent by submerging the plumb bobs in a heavy liquid, such as heavy lubricating oil.

Cylinder liner replacement. Nothing has been said as yet about the cylinder liner, which will now be considered. Liners

are replaced more often than bearings and many other parts. Therefore the chances of having a liner misaligned are more numerous, owing to improper installation or poor construction. Some maintenance men seldom check a replaced liner to see whether it is perpendicular to the crankshaft. A check of this nature is worth while. Excessive liner wear and piston sticking may result from misalignment. Neglecting to check alignment may prove expensive.

One method of checking a replaced liner is to place a straight edge and a level across the top of the liner. Another method is shown in Fig. 6-7. The rod or stick from which the weights or bobs are suspended by fine wire is placed across the top of the liner parallel to the shaft. The weights are suspended through the liner to the shaft and about two or three inches from the liner wall. The distances O and P should be equal and, as a check, M and N should also be equal. The rod is now placed perpendicular to the shaft and the procedure is repeated. This method, if accurately performed, will give a definite indication regarding the alignment of the liner.

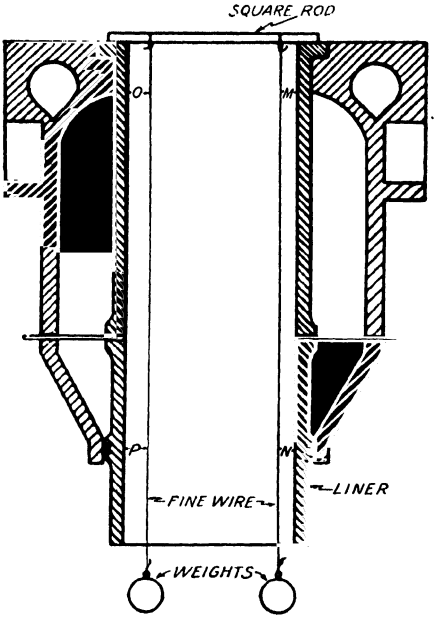


FIG. 6-7. Checking a replaced liner.

The problem of alignment involves not only the items discussed but also practically all moving parts throughout the engine. Only the more complicated procedures are brought out here for instruction purposes, since too much space would be required in discussing in detail all necessary alignments.

Bearing clearances. While some maintenance men give too little attention to the subject of bearing clearances, it is one of the important factors in operating Diesel engines. It is good practice to allow standardized bearing clearances that are applicable to the particular type of engine. These clearances

have been standardized by experience and testing in the field as well as the experimental shop. It is just as detrimental to have too much clearance as not enough because the bearings will pound out. When setting bearing clearances, one should be careful to allow only what is considered standard tolerance. A table of bearing clearances is shown here to indicate the extent and importance of this information. The operator of any engine should obtain from his engine builder precise information and make use of it every time he makes any changes on the engine.

Methods of checking bearing clearances. There are several ways of checking bearing clearances. One simple method is the use of a thickness gauge, when possible, such as on pinion and main shaft bearings. Another method is known as "leading" bearings, not applicable to the high-speed, precision type of bearing. Leading is simple, takes more time than other methods, but has been found satisfactory when properly done. In leading bearings, the caps are removed and a lead of fuse wire is laid over the crankshaft's journal at each end of the bearing, the ends of the lead wire about one inch from the shims.

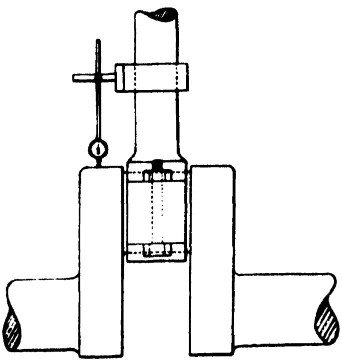


FIG. 6-8. Checking crankpin bearing clearance.

The bearing cap is then replaced and tightened. It is then removed and the lead wire is "miked" for thickness. The measure of the thickness is the measure of the clearance, unless the wire was imbedded improperly into the babbitt surface, which occurs when a large wire is used on soft metal. In checking crank bearing clearances, the crankpin is placed near the top dead center position and the bearing is allowed to be lowered enough to insert lead wire in the bottom cap, one at each end. The cap is then tightened to the rod with proper tension of the bolts. The cap is removed, the lead wire taken out and measured for thickness. The clearance of the bearing is indicated by the thickness of the wire. The size of the wire to be used depends upon the size of the bearing and the required tolerance. A two-ampere wire is satisfactory for leading bearings on the average engine.

Another method of checking crankpin bearing clearances is shown in Fig. 6-8. The dial indicator is clamped to the connecting rod with a dial pin resting on the crank web. After the dial is set and the initial reading is noted, by means of a long rod or timer, the connecting rod and bearing are forced up to the crankpin. The dial reading is again observed and the difference between this reading and the initial reading is the bearing clearance. The rod may be reversed and the reading repeated.

Crankshaft inspection. The crankshaft is one of the most important and costly items in an engine. As its construction is complicated, the replacement of the shaft would mean a long period of shutdown; therefore, it must be given careful attention and care with respect to alignment. Failure of crankshafts is attributed largely to misalignment caused by high and low bearings, which permit the shaft to "bow" up and down when revolving. This sets up a vibration and causes "flectures," which eventually fracture. Realignment of the shaft periodically is a good practice.

Fractures of shafts. Fractures are usually found in the fillet of the journals near the crank web, as a result of concentration of stresses at these points. Such cracks are seldom visible to the naked eye in the early stages; consequently some procedure must be followed to detect these possible fractures before the shaft is completely broken. The following is one of the several steps that may be taken:

Clean the shaft until it is free of all greases and oils. After drying thoroughly, apply a thick coat of a mixture of alcohol and powdered chalk around the journal at the fillets. Allow mixture to dry.

The alcohol will dry quickly, leaving the surface coated with a thin coat of chalk. As the shaft is rotated, any minute fractures that may be present will be indicated by streaks of oil working out of the crack and appearing in the white chalk as a stained line showing the extent and location of the fracture. For the best results, the shaft, when rotated, should be suspended by only the two end bearings.

The crankshaft must be straight and supported equally by all bearings with equal load on them, otherwise it will "bow," and fractures will occur after a sufficient length of time of operating in this condition. When a shaft is checked for

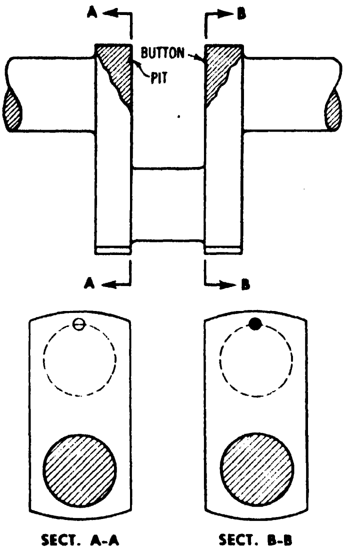


FIG. 6-9. Checking the shaft for flecture, or "bow."

fractures or "bow," the crank webs should be prepared to receive micrometer calipers or a strain gauge.

Use of the strain gauge. Make small center punches in the webs, preparing the indentations to receive the points of the strain gauge. Where micrometers are to be used, the webs should have a pit on one side and a button on the other near the top and center of the web, as shown in Fig. 6-9. The pit is made by hammering on a small steel ball held against the web with an improvised clamp. The indentations made by the ball will receive the stem of the micrometers. Drilling to the depth of the taper of a small drill will also make a suitable pit. The button that contacts the head of the clamp calipers is made by a specially machined tool. See Fig. 6-10 (a) and (b). Another method of making the button is shown in Fig. 6-10 (a). A hole the size of a small steel ball is drilled into the web to a depth equal to $\frac{3}{4}$ of the ball diameter. This ball is inserted in this hole and the metal calked around it to prevent its falling out. The pit and button must be smooth and free from burrs and carbon deposits so that one can obtain correct micrometer readings between the two points.

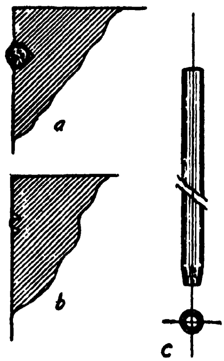


FIG. 6-10. Method of making button on crank web for use of micrometer calipers.

Readings are taken at four positions of the crankpin, namely, top, back, bottom, and front, designated as *A*, *B*, *C*, and *D*, taken in the order of rotation of the engine. If the readings are equal in the four positions of the crankpin, the indications are that the shaft is straight. Variations in these readings will be indications that the shaft is misaligned, as a result of bends or misaligned bearings.

In Fig. 6-11 is shown the greatly exaggerated position of a shaft caused by a low bearing. In this case, the reading, when the crankpin is at the top or in A position, will be greater than when the crankpin is at the bottom or C position. In the event that the connecting rod is in place when the readings are taken, the crankpin can be slightly off bottom dead center when the C reading is taken, so that the strain gauge or micrometer will clear the rod. A difference in reading when the crank is in the front or D position or in the back or B position will indicate possible horizontal misalign-

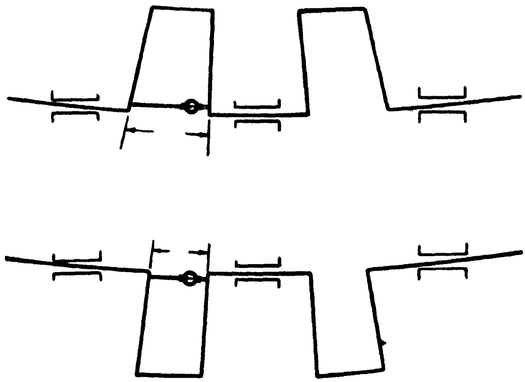


FIG. 6-11. Exaggerated position of a shaft caused by a low bearing.

ment of the bearings. A shaft in this position must be corrected, as it causes a reversing of the fiber stresses in every revolution of the shaft. The magnitude of the fiber stress can be determined by the formula:

$$P = \frac{43{,}500A \times T}{L^2},$$

where P equals stress in pounds per square inch, A equals maximum deflection in thousandths of an inch, T equals thickness of the crank web in inches, and L equals one half of the stroke of the crank. This formula is Hooke's law as applied to this type of beam.

It is good practice to make sure that a shaft is being supported by all of its bearings when in operation. Shafts, owing to their stiffness, will show to be in perfect alignment, yet when put into operation under heavy load may be "bowed." Bridge

gauge readings will readily indicate whether or not the shaft will bow under loaded conditions.

Taking bridge gauge readings. Fig. 6-12 shows one method of taking gauge readings. First the crank adjacent to the main bearing journals being gauged is placed in a horizontal position. The gauge is set over the journal so that the micrometer pin is directly over the center of the shaft. The micrometer is adjusted until the pin touches the shaft, and then the reading is observed. By means of a small jack placed under the crank web and against a timber placed in the frame the main journal

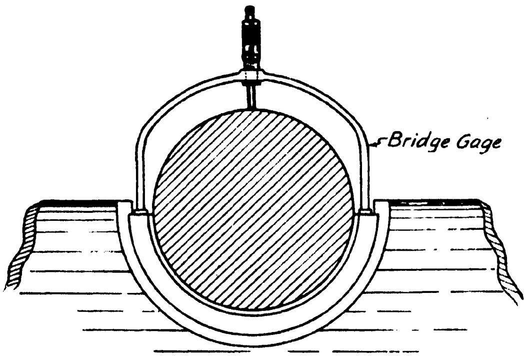


FIG. 6-12. One method of taking bridge gauge readings.

is forced downward. The micrometer is again adjusted to touch the journal and the reading is noted. If the two readings differ, the indications are that the bearing is low in respect to the other bearings. If the second reading is the same as the first, the journal is being correctly supported by its respective bearings. This procedure is repeated on all main bearings and journals. All bearings found to be out of alignment should be corrected, as such conditions may lead to serious trouble, such as crankshaft failure. Main bearing shells do not always fit the bore of the bedplate of an engine. These should be checked with "bluing" and all lapped into place. A level on the main journals will indicate whether or not the shaft is level with respect to the bedplate when the engine is sitting level.

Crankpin bolts. The failure of a crankpin bolt can cause great damage to an engine. These failures are usually the result of one of several causes: (1) bolts tightened too much, (2) bolts tightened too little, (3) bolts made of inferior steel,

(4) bolts not designed for sufficient strength, or (5) bolts in service too long and weakened by fatigue. Crankpin bolts should be of the best alloy steel with the right heat treatment. Engine manufacturers are now alert to the importance of furnishing satisfactory bolts. Therefore, the operator can forget items (3) and (4) if the bolts have been purchased from the engine manufacturer. These questions are beyond the control of the operator. The practice of buying replacement bolts from a local machine shop is questionable. These shops rarely have the facilities for heat-treating the bolts and may furnish bolts of any steel in stock.

Several rules have been advanced for the operator's guidance in tightening crankpin bolts. Those which specify a certain length of wrench and a certain pull at the end are questionable and should be used with caution. Differences in the thread friction of the nut will set up different bolt tensions with the same pull—and even then, the amount of pull is hard to gauge. The accepted practice is to caliper the bolt length before any strain is put on it, then tighten up on the nut until there is a certain elongation. The manufacturer can recommend the proper elongation for any bolt of his make. If the tension on the bolt is not enough, the box will slam in operation; if the bolt is tightened too much, it will be weakened and fail from fatigue strains. To make sure that the over-all measurements will be strictly comparable, there should be center-punch marks on the head and opposite end. Some bolt manufacturers drill the ends and insert special pins for such measurements.

Checking crankpin bolts. When a crankpin box is disassembled, the bolts should be examined closely for fine cracks that might indicate fatigue. Bolts should be examined every 8000 to 10,000 hr of operation. One manufacturer recommends scribing tram marks on the shank of the bolt near the head and on the thread near the end opposite, so that the length of the bolt can be checked against the original length to determine whether there has been any elongation. This does not complete the inspection, however, in view of the fact that any scoring or breaking of the surface of the bolt is likely to start a weakness. A comparison of the over-all length, as recommended in tightening, will not be enough, for the bolt end or head may have been worn between measurements. The complete and most reliable procedure is to tighten the bolts

correctly, as checked by elongation, inspect for cracks and imperfections periodically, and renew at definite intervals.

Roundness of pins. When new bearings are being fitted, pins should be calipered for roundness. Irregularity of as much as 0.0015 in. per inch of diameter should be corrected. It is preferable to discard the worn piston pin and replace it with a new one. Such wear in a crankpin would involve the entire main shaft, and it is necessary to grind the pin down to perfect roundness which then requires undersized bearings. In regrinding, care should be taken to see that sharp corners are avoided

REPORT OF TEARDOWN INSPECTION

Cylinder No.__________ Date__________

		Main Bearing Measurements		Connecting Rod Bearing Measurements	
		Top Half Shell	*Bottom Half Shell*	*Top Half*	*Bottom Half*
Inboard	F				
	A				
Crown	F				
	A				
Outboard	F				
	A				

PISTON and CYLINDER

	Piston		Cylinder		Piston Pin
Top	F	A	F	A	
Center	F	A	F	A	
Bottom	F	A	F	A	

PISTON RING MEASUREMENTS

Ring	Side Clearance	End Gap	Groove Depth Clearance	Replace
1				
2				
3				
4				
5				

REPORT OF TEARDOWN INSPECTION—2
Remarks on Inspection and Repairs

	Condition*	Repair or Replacement†
Intake valve		
Exhaust valve		
Air-starting valve		
Safety valve		
Fuel injection valve		
Cylinder head clearance		
Connecting rod, centering of top		

Inspected by________ Hours since last overhaul________

Measurements: All measurements with micrometer in thousandths of inch. Check all measurements with dimensions "worn" in Wear Limit Table.

* Condition: Warped, pitted, burned, broken, etc.
† Repair or replacement: Refaced, ground, replaced, new.

between the pin and the crank cheeks. These sharp corners weaken the shaft and may start cracks. That is why pins and journals are terminated with rounded fillets. *Do not file faces of bearing caps.*

Checking alignment in place. The connecting rod alignment can be checked without removing the rod from the engine, by measuring the side clearance of the crankpin box at different crank positions. Any misalignment will cause the box to bind against one cheek or against both alternately.

Adjusting piston to cylinder head clearance. The clearance between the piston and the cylinder head can be adjusted by adding or removing shims between the foot of a marine-type rod and the top-half bearing box. When a cap-type rod is used, the only possible adjustment is changing the copper cylinder head gasket to a thicker or a thinner gasket. To check the clearance, a soft lead wire should be placed on the piston top so as to come between the piston and head where the clearance is minimum, or at a point for which the manufacturer has specified what the clearance should be. The wire can be inserted through the valve cage openings. If there are no cages, or if the wire cannot be inserted in that way, the head can be removed, the wire placed, and the head reassembled. If the correct clearance is larger than the largest size of lead wire available, a rope of lead wires can be tried. After the wire

is placed, the engine should be barred over, the wire removed and micrometered. The compressed thickness will equal the clearance. Shims are inserted at the foot of the rod to decrease clearance, and vice versa. A thinner gasket is substituted to

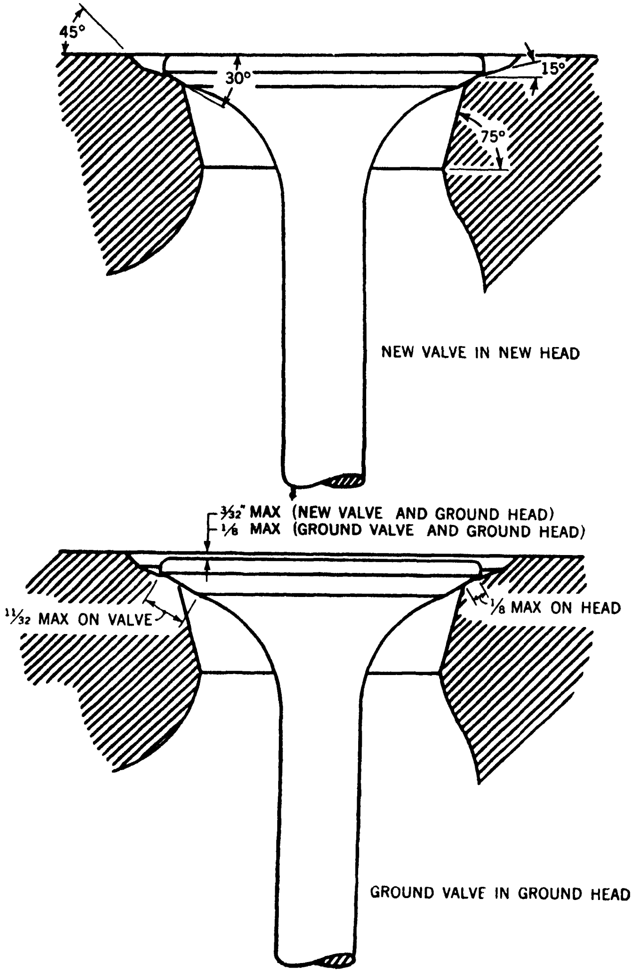


FIG. 6-13. Exhaust valve clearances.

decrease the clearance. If after correcting the clearance to the amount specified the compression is still not correct, look for leaking piston rings or poorly seated valves. After an adjustment that decreases clearance, bar the engine over slowly several times to make sure the piston is clear of the head.

PREVENTIVE MAINTENANCE MONTHLY PROGRAM or Every 600 Hours

1. Clean lubricating oil strainer.
2. Clean lubricating oil filters.
3. Inspect condition of lubricating oil.
4. Clean fuel oil filters.
5. Add grease to fittings.
6. Check overspeed shutdown device.
7. Drain moisture from fuel pressure tank or day tank.
8. Check condition of zinc plates in heat exchanger and oil cooler.
9. Drain water fuel storage tank.
10. Remove cylinder head covers.
11. Take peak pressures.
12. Take compression pressures.

QUESTIONS

1. The wear limit table (Table 6-1) lists eight groups of engine parts or elements for which dimensions, clearances, and wear limits are given. Make a list of the kind and size of measuring tools, such as micrometers and gauges, required for checking an engine of this kind.

2. What special tools, in addition to wrenches, would be required for teardown, repair, and reassembly of an engine of this kind?

3. Inspection report forms for main bearings, piston and cylinder, piston ring measurements, and valves for use in making a top overhaul are given on pages 194 and 195. Make up similar report forms for other groups of engine parts.

4. Refer to Chapters 8, 9 and 10 for report forms used for making records of inspections of engine deposits, and so on. Study the possibility of making up report forms for recording other information and data, including parts replacement, and so on.

5. Students of Diesel operation and maintenance should have actual experience and training in making all measurements listed for a particular engine, devising forms, recording the information, and calculating on graphs the rate of wear against hours of operation.

6. A major part of any laboratory or shop training course should include teardown and disassembly, inspection of every part, measurement of all dimensions and clearances, and the calculation of all wear found. If you are operating an engine, get the maximum benefit from your next experience with overhaul, by systematizing your methods and records.

7. Practical operators who study this book will understand their own instruction book in a new light. Check the items included in your instruction book, with the layout of maintenance procedure, and the inspection procedure in this and Chapter 8. Check your tools and equipment, including instruments. Do you possess sufficient tools to carry on maintenance work?

8. How is the wear in the rocker arm bore checked; and how is the wear corrected?

9. What causes flat surfaces on the cam rollers?

10. What is used to measure the wear between roller and pin?

11. When valves are found to be leaking, what are some of the causes to be looked for and determined?

12. How and with what is the seat remachined?

13. What condition of the valves require refacing?

14. What can be done if valve springs are found to be weak?

15. When valve cages are being installed, what precaution must be taken with the bolts? If not taken, what may be the result of uneven tightening of the bolts.

16. What is the method recommended for checking bearing alignment, as shown in Fig. 6-1?

17. How is the parallelism of bearing bores checked relative to each other?

18. Why is it important that the alignment of the piston be checked?

19. What may happen when the cylinder liner is installed with poor or improper alignment? How is the alignment of the liner checked?

20. A lead wire is used for checking bearing clearances on large engines. How is the lead wire measured for thickness? What is used for this purpose? How accurate should it be read?

21. Describe the usual method of inspecting the crankshaft. How are the chalk and alcohol used for this purpose?

22. What is the strain gauge? Describe it use.

23. If there is reason to believe that a crankshaft is "bowed" during operation under heavy load, what kind of gauge may be used to indicate whether the shaft will bow under such conditions?

24. What is the bridge gauge used for? Is it used to check a bearing believed to be low?

25. What are the five factors that may contribute to failure of crankpin bolts, or connecting rod bolts?

26. What is considered the most reliable method of checking crankpin bolts?

27. What is used to check the length and elongation of the bolts?

28. At what intervals does the manufacturer of your engine recommend the replacement of the bolts? What elongation does he recommend? Is this information found in the instruction book?

29. Should the face of precision bearing caps ever be filed?

30. What method is used to measure the cylinder head to piston clearance?

31. Should the compression be checked after this clearance is adjusted?

32. How is the piston to cylinder head clearance measured on the engines with which you are familiar?

33. On what type of engine can the clearance be regulated by removing or adding shims to the foot of the connecting rod?

34. What is meant by connecting rod bolt torque?

35. What is a torque wrench, and how is it used?

36. What is meant by "cylinder head to block nut torque" in the table of wear and dimensions?

37. Look at Fig. 6-13, and list the exhaust valve clearances to be checked. What tools and methods are used?

CHAPTER 7

INSPECTION PROCEDURE

Introduction. The use of a definite inspection procedure is an essential feature of maintenance work. After a suitable maintenance policy has been adopted and proper equipment and facilities provided, a standard inspection procedure should be put into effect. Inspection methods and practices evolved by experience and made standard by the Diesel engine manufacturers are essential to securing low-cost power from the modern Diesel engine. Intelligent inspection of engine parts during the teardown and reassembly of the engine is the basis of this practice.

Maintenance and inspection work differ only in the amount of reconditioning and repair work done on the engine elements and parts. The inspection of the engine and its parts, whether stationary, marine, locomotive, portable, or automotive, comprises essentially the same procedure. The special equipment required is related to the maintenance work and the type of engine. Experience determines the time between various maintenance operations and overhaul periods. A considerable percentage of the time involved in inspection and maintenance of the engine is consumed in removing parts or dismantling to gain access to various parts to be repaired or inspected. As pointed out in the previous chapter, a method of insuring a complete and thorough inspection justifies the work and effort involved. Since all the details of inspection and maintenance would require lengthy discussion, a condensed summary of inspection and maintenance procedure is herein presented, to include teardown and reassembly of the entire engine.

Inspection of the main engine. Certain inspections are made before the overhaul starts, some while the engine is running, and others with the engine shut down as conditions

require. The steps to be taken are as set forth in the following list:

1. Indicate the engine, using a pressure indicator such as the Premax. When possible, indicate and study the compression and firing pressures. Any irregularities in the pressures developed during the previous period of operation may be determined. A record is made of the data indicated.

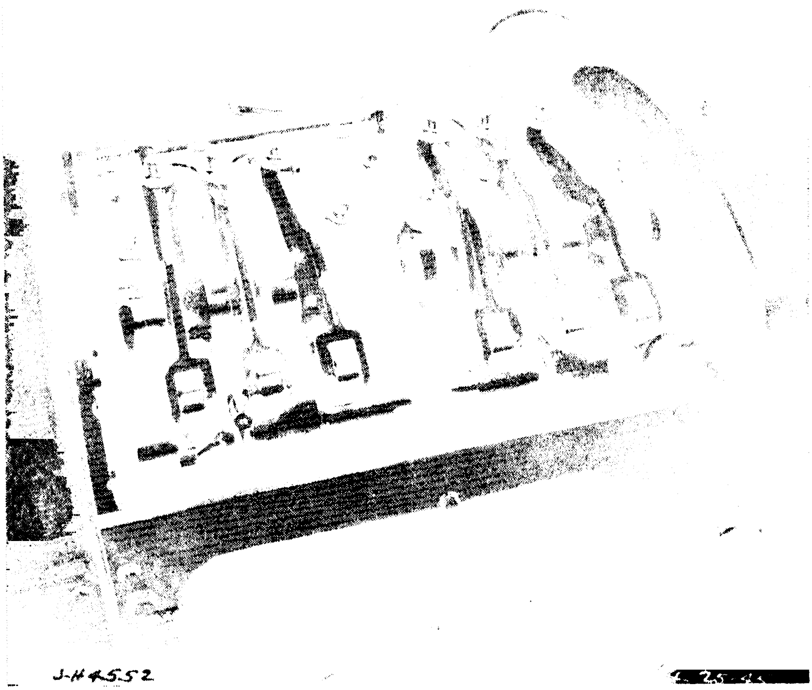


FIG. 7-1. Close-up of cylinder head assembly, showing camshaft and rocker arm assembly. Camshaft fitted with forged, hardened, and polished steel cams, three per cylinder, an intake, an exhaust, and one cam for performing two functions—namely, fuel injection and air starting. Push rods and tappets are eliminated by the use of the overhead camshaft, with the rocker arms bearing directly on the cams. This is an ideal design for unit injection systems.

2. Check the pressures of the water, lubricating oil, fuel oil, and air compressors, if air injection.

3. Inspect all relief valves. Test lifting pressure in the pressure relief valves.

4. Check valve timing and valve lift.

5. Observe governor action and performance.

6. Check the fuel pump for leaks in plungers, and so forth.
7. Make every effort to locate abnormal noises.
8. Investigate any condition not in keeping with good practice, as indicated in previous chapters.

Removal and inspection of parts. Secure proper tools and equipment, and proceed as follows:

1. Remove rocker arms and valve mechanisms.
   a. Examine for flat spots and wear on rollers, pins, and bushings.
   b. Note wear on tappets.
   c. Check fulcrum pins and bushings for wear.
2. Remove valves from head and prepare to recondition.
   a. Clean and grind until all pits disappear. If valves or seats have warped or are badly pitted, it may be necessary to reface.
   b. Inspect valve springs for length and tension.
   c. Inspect valves carefully to determine wear and general condition.
   d. Inspect all moving parts, including valve stems, valve stem bushings, guides, lock washers, and so on.

3. Remove main cylinder heads after determining head clearance with a lead wire or copper pipe.
   a. When convenient, remove exhaust and intake headers; when not convenient, block in place.
   b. Clean the heads. When necessary, use an approved scale remover.
   c. Remove all clean-out plates and plugs to facilitate cleaning.
   d. Examine carefully for defects, such as cracks, leaks, and bad seats.
   e. Replace clean-out plugs and plates.
   f. Checks should be made for cracks, erosion, and so on.

4. Wash and clean engine jackets, exhaust header, air intake system, and such members.
   a. If scale is present, use approved scale remover.
   b. Determine the thickness of the scale deposit.
   c. Remove the scale remover acid by flushing out with soda solution.
   d. Determine the kind of scale deposit.

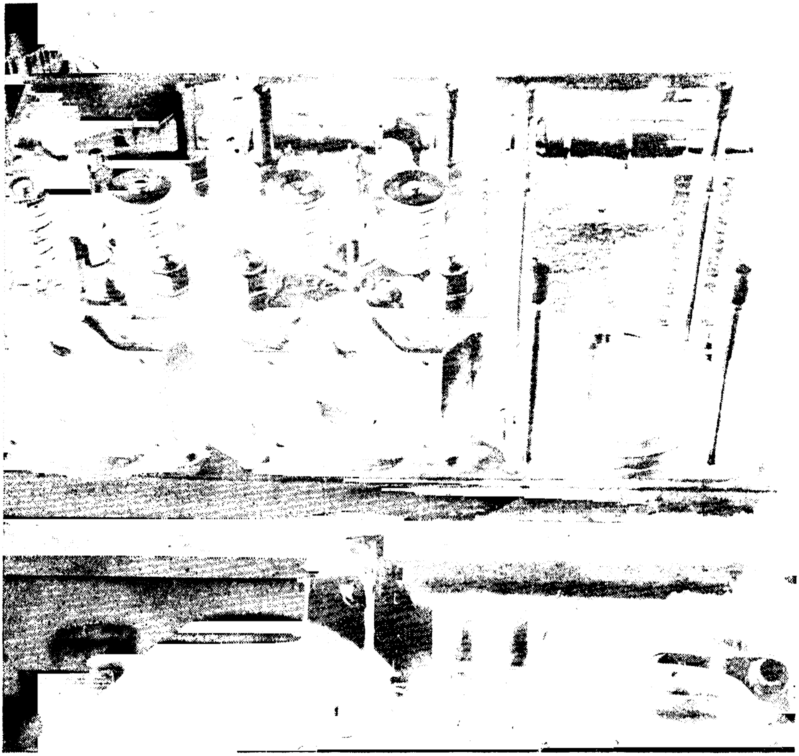


FIG 7-2. Close-up view of top of assembly with rocker arms removed, showing cylinder heads. The assembly is cast iron and is designed for maximum rigidity and cooling The removable cylinder liners, the cylinder head studs, and the method of removing the heads are also shown.

5. Remove crankcase doors and casings over camshaft drive and other auxiliary drives.

   a. Wash out crankcase with kerosene or fuel oil, hot water, or steam.
   b. Break connections on end of the engine to facilitate work.
   c. Replace as soon as cleaning is completed.

6. Block connecting rod bearings. Proceed as follows and make careful record of data:

   a. Measure connecting rod bolt length before relieving strain on the nut.
   b. Relieve strain on the nut and remeasure the bolt length.

c. Tighten set screws for holding connecting rods bolts in place and remove nuts.
d. Compare measurements made in (a) above with previous measurements made when bolts were installed. Compare measurements in (b) with original bolt lengths.

7. Pull pistons, one at a time or all at once, as convenient, making use of proper equipment.

a. Remove rings and note carefully general condition and any indication of blow-by past the rings.
b. Clean and inspect piston inside and out for cracks, scores, and other defects.
c. Remove piston pins, measure, and note dimensions in planes perpendicular and parallel to piston pins.
d. Examine piston pin bearings and adjust to proper clearance.
e. Use surface plate to obtain check as to whether piston bearing is true in planes perpendicular and parallel with crankpin bearings.
f. Thoroughly clean oil holes in connecting rods and insert corks in the holes until rods are installed.
g. Install piston pins in piston after measurement of wear.
h. Measure piston skirt. Use wood block and sledge to adjust piston skirt dimensions so that skirt diameter is approximately .0015 in. smaller parallel to pin than perpendicular to it.
i. When oil-cooled pistons are being inspected, break out oil-cooling header as much as possible and clean.

8. Remove connecting rod bearings from crankcase. Use proper method as previously outlined.

a. Inspect carefully for loose babbitt, cracks, and other defects. Measure for wear.
b. Inspect connecting rod bolts very carefully. Note the bearing surfaces under the head and evidence of damage.
c. Suspend connecting rod bolts from a string, nonmetallic, and sound for cracks.
d. Have bolts magnafluxed when facilities are available.

9. Measure with micrometers the distance between crank webs.

a. Take measurements on all cranks in the four quarter positions as previously instructed.
b. Use strain gauge when available and indicated.

10. Wear limit check.

a. Check measurements against wear limits.
b. Make careful records of all measurements.

11. Remove main bearing caps. Inspect bolts in the same manner as for engine connecting rod bearings.

a. Inspect for cracks, loose babbitt, and evidence of heavy bearing areas, wiping of bearing metal, scores, and cut shafts.

12. Take bridge gauge readings of all main bearings, as indicated in foregoing chapters.

a. If bridge gauge is not available, spot frame directly above journal and measure distance between spots and shafts.
b. Press shaft into shell with either clamp or jack and take second bridge gauge reading. Bridge gauge readings before and after pressing down of main bearings should not vary more than 0.002 in.
c. In case of deflection that is more than 0.002 in. in any one direction, as shown in item 9, remeasure the distance between the crank webs at the point in question while the crankshaft is clamped in the bearing.
d. Examine the crankshaft for cracks. (Paint with alcohol and chalk.)

13. With an approved shafting level, determine position of all main journals.

a. Determine pitch per foot and direction of slope.

14. Roll bearings of engine out of sockets, inspect for defects, look for evidence of wear and relative pressures.

a. Compare thickness when wear is indicated.
b. Determine if any bearings are wiped.

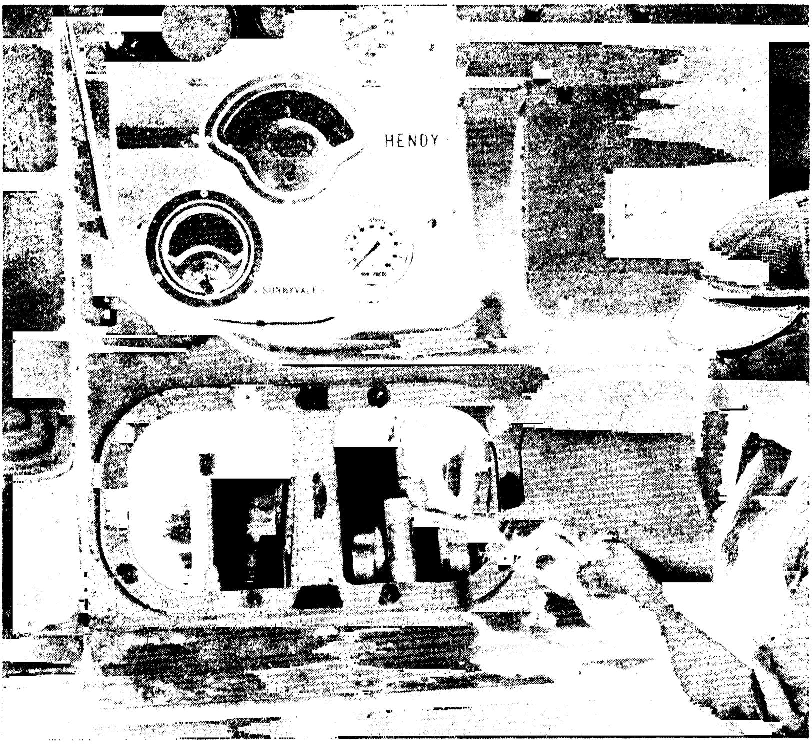


FIG. 7-3. Large ports in the side of the engine lower frame of the Hendy Series 20 engine make the crankshaft accessible for inspection and adjustment.

15. From items 11 to 14 above, determine the cause of any crankshaft misalignment. A crude sketch of shaft showing the journal level and web distances will aid in finding the cause of misalignment.

a. Leave crankshaft with no more than 0.002-in. difference in measurements between webs in any two diametrically opposite positions.
b. Check strain gauge data, if obtained.

16. Replace main bearing caps and lower shells, giving proper clearance after oil passages have been thoroughly cleaned.

a. Clean oil passages and blow piping with air pressure.
b. Take leads on the bearings with original shim.
c. With this information, determine amount of shims to be removed to give proper clearance.

d. The rule generally followed for main bearings clearance is to allow 0.0005-in. clearance per inch of shaft diameter, and to this amount add 0.002 in. Thus, a 10-in. shaft would require 0.007-in. clearance.
e. Use precautions to insure the pulling down of caps evenly and squarely with main engine frame.
f. Check crankshaft and thrust, and as far as possible adjust to builder's recommendations.

17. Inspect vertical shaft and shaft drive, or other camshift drives, and the like.

a. Remove all housing necessary.
b. Inspect and adjust bearing clearance and thrust.
c. Note gear clearance and wear, and make a record.
d. Inspect timing chain and pinion for tension, wear, and other defects; inspect timing gears, if any.

18. Examine the camshaft.

a. Adjust bearing clearances and thrust.
b. See that all cams are tight and in their proper position.
c. Examine the cams for excessive wear.

19. Check fuel pump, fuel system piping, fuel pump for wear in drive, and so on.

a. Check all moving parts, such as pins, bushings, crossheads, links, straps, eccentrics for excessive wear, adjustment, and clearances. Look for any defects.
b. Check and repair all pumps; check and try valves, removing valve bodies when necessary.
c. Reset governor valves to builder's standards.
d. Make sure all joints, bearing races, as well as other moving parts, are inspected for breakage, wear, and lost motion.
e. Check all valves for required spring tension.
f. See that all valves have proper lift.

20. Inspect the governor.

a. Dismantle the governor and completely overhaul it when this is necessary.
b. Note all joints and bearing races, checking moving parts for wear.

c. Check all nuts, keys, and the like, and see that they are in the right place and tight.
d. Check all parts to see that they work.

21. Replace housing on such items as camshafts driving mechanisms. Examine for cracks.

22. Examine and inspect lubricating oil pump and elements of pump wear.

a. Dismantle lubricating oil pump. Observe the general conditions and compare with requirements for satisfactory operation.
b. Inspect bearing and shaft conditions of drive.
c. Inspect and renew any packing, and so on.
d. Note condition of and repair driving mechanism when needed.
e. Completely dismantle, clean, and repair relief or by-pass valve.

23. Measure liner diameter with inside micrometers, and examine the liner surfaces for scores and wear.

a. Take the measurements both perpendicular and parallel to the crankpin.
b. Measurements in position to be taken near the center of the liner above the oil hole, about one inch from top of ring travel, using standard locating iron for determining exact position.
c. If abnormal conditions are found, take other measurements as shown in the report form in the previous chapter.
d. Operate force feed lubricator by hand and thoroughly check all lubricating oil lines from the lubricator to the cylinder. Special attention to the lubricator heads in the cylinders is required.

24. Replace connecting rod bearings. They should have been cleaned, inspected, and the shells also should have been inspected.

a. Properly place leads and original shims so that the clearance can be determined later.
b. Again carefully inspect connecting rod bolts, place in position, and secure with set screws provided for that purpose.
c. Replace with required compression shims.

25. Prepare piston and install. The rings should be checked carefully, as previously explained.

a. Fit the rings in the liner and determine the gap, which may be 0.005 in. per inch diameter of the ring.
b. Determine width and condition of the ring groove in the piston.
c. Install the rings, making sure of proper side clearance.
d. Thoroughly oil the liner, piston, spacer rings, and insert the assembly properly.

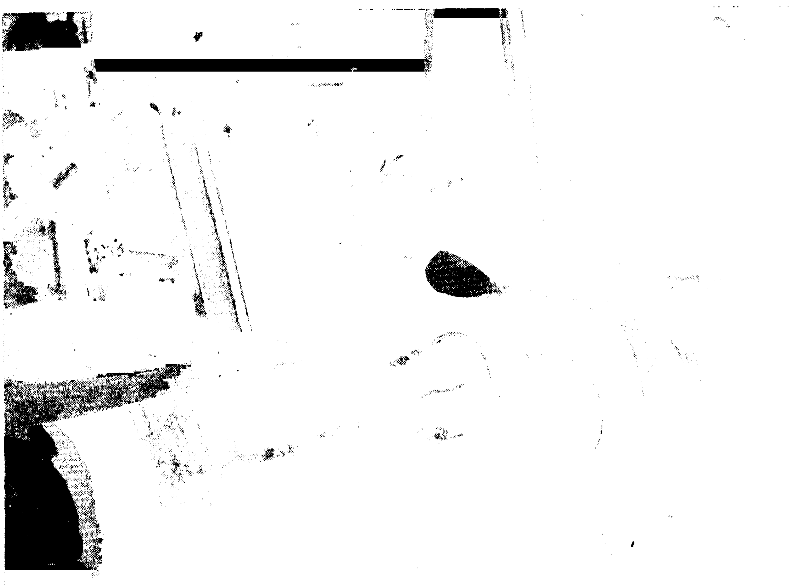


Fig. 7-4. Inspecting crankshaft for cracks in the fillets, showing use of powdered chalk solution for this purpose.

26. Adjust connecting rod and bearing.

a. With the piston and connecting rod in place, tighten bearing bolts to insure pinching of lead wire to actual bearing clearance.
b. Remove lead wire to determine original clearance and adjust shim thickness to give proper clearance.
c. Tighten bolts in accordance with standard torque tables for this engine. Use a torque wrench as instructed.

d. Operate hand lubricating oil pump to see that oil runs freely, through all bearings. Check oil passages or piping for leaks.

27. Replace cylinder heads. Lift heads with proper equipment and do not damage.

a. Make sure that all contact parts are clean and gaskets are annealed before reusing.
b. Tighten all cylinder head bolts evenly with equal tension; use torque wrench when available.
c. Check head clearance.

28. Replace valves in the head, if cage type.

a. Make sure that all copper gaskets are annealed and cleaned.
b. Clean seats and all contacting surfaces, making sure that all seats are free of foreign matter.
c. Tighten valve or cage bolts evenly and gradually, and allow for expansion. Use wrench without cheater for this work.

29. Replace rocker arms.

a. Allow recommended roller clearance for all valves.
b. Check push rod and cam clearance.

30. Remove compressor valves.

a. Clean and grind.
b. Check valve lift.
c. Examine all parts for excessive lubricating oil consumption.

31. Measure head clearance, air compressor.

a. Use lead for this purpose.
b. Examine rings of compression piston.

32. Remove cylinder head and pull compression piston.

a. Clean and observe carbon deposits.
b. Inspect rings for wear and defects; replace worn rings.
c. Examine and adjust pin bearing and pin.
d. Examine compressor liners, taking measurements with micrometers in the same manner as with power cylinders.
e. Replace compressor piston.

f. Examine and adjust connecting rod bearings, making proper allowance for head clearance.
g. Inspect all air piping and air bottles; clean with steam or air or by any other approved method.

33. Inspect and replace compressor head and valves.

a. Take same precautions as with power cylinders.
b. Check all clearances.

34. Set timing of all valves.

a. With bevel contractor or tram, check opening and closing of exhaust valves and inlet valves on one cylinder.
b. Use dial indicator or bevel protractor or tram to set opening and closing of fuel valves (on air injection engines).
c. Thoroughly inspect crankcase for rags, blocks, and other loose objects; have this rechecked by another person.
d. Replace crankcase doors and other parts.

35. Check foundation and frame bolts and alignment of the engine bed plate.

36. Examine flywheel bolts and see that the nuts are tight.

37. Prepare the engine for starting.

a. Operate mechanical lubricator by hand to supply oil to the pistons.
b. Bar the engine over several revolutions and spot for checking.
c. Check air pressure in starting air tanks or bottles.
d. See that all tools and other materials are out of the way of the moving parts of the engine.
e. Take all necessary steps common to starting the engine.

38. Start the engine.

a. The operating engineer should start the engine.
b. The machinist should not touch the gate valves, switches, or other apparatus incident to the operation unless the engineer is present.
c. The engine should be shut down, doors removed, and an inspection made within ten minutes after first starting an engine after an overhaul.

d. Inspect thirty minutes after the first stop. Then run two hours at full load and inspect. Then run four hours and inspect. If the inspection is satisfactory to the engineer and to the mechanic, the engine should be ready to place in service.

39. Records. Records of measurements, reports of conditions found during the teardown inspection, and adjustments made when the engine is assembled should be made and kept for future reference.

Since the condition of the engine found by inspection is the reason for inspection, the interpretation of the data on these observations must be made. The meaning and significance attached to the various conditions revealed by inspection is subject to further discussion in chapters that follow. This will cover piston rings, piston and cylinder, major engine bearings, and other vital parts of the engine.

Clearances, dimensions, and wear limits. The table of Clearances, Dimensions, and Wear Limits, Table 7-1, is a carefully prepared list of items that are measured when a typical marine engine is torn down for inspection. It is given here to illustrate the various measurements that must be made.

This information is essential to any intelligent maintenance and inspection program. It is an illustration of the kind of information required and the way it should be listed and kept available. The manufacturer's instruction book usually contains this information together with instructions on making the various measurements. *Only a study of the instruction book of a particular engine makes it possible for the operator to perform this important duty.*

Inspection of auxiliaries. The Diesel plant includes essential auxiliaries that must be regularly inspected. There are several auxiliary systems that control the economics of the Diesel engine. The inherent characteristic of each type of Diesel engine makes a definite demand upon the auxiliary system's equipment. Through a regular system of maintenance and inspection and a study of the records thus obtained it is possible to know the internal condition of the engine within fixed limits. With this knowledge of the reasonable working limitations of the engine, replacements and adjustments can be provided for well in advance so as to eliminate aggravated difficulties. The corrosive properties of fuel oil, the scale-forming tendency of

the cooling water, and the deterioration of the lubrication require diligence and permits of no neglect in keeping up the equipment.

The schedule here given does not list the maximum allowable time lapse between specific maintenance procedure suggested—but such data should be added for each individual installation after consultation with the engine builder, the accumulation of experience, and information from the manufacturers of the

TABLE 7-1

CLEARANCES, DIMENSIONS, AND WEAR LIMITS OF A TYPICAL MARINE ENGINE

MAIN BEARINGS	
Crankshaft diameter (new)	7.2485–7.2495 in
Shell to shaft clearance (new)	.006 –.009 in.
Shell to shaft clearance (max. allow. worn)	.030 in.
Shell thickness (new)	.3715–.37275 in.
Shell thickness (minimum allow. worn)	.360 in.[1]
Thrust bearing end clearance (new)	.030 –.035 in.
Thrust brg. end clearance (max. allow. worn)	.118 –.125 in.[2]
CONNECTING ROD	
Crankpin diameter (new)	6.2485–6.2495 in.
Shell to shaft clearance (new)	.0065–.0085 in.
Shell to shaft clearance (max. allow. worn)	.025 in.
Shell thickness (new)	.246 –.24675 in.
Shell thickness (min. allow. worn)	.238 in.[1]
Piston pin diameter (new)	2.999 –3.000 in.
Piston pin diameter (min. allow. worn)	2.992 in.
Piston pin bushing outer diameter (new)	3.872 –3.873 in.
Piston pin bushing outer diameter (min. allow. worn)	3.864 in.
Piston pin bushing inner diameter (new)	3.0025–3.0035 in.
Piston pin bushing inner diameter (max. allow. worn)	3.012 in.
Piston pin to conn. rod bushing clearance (new)	.0025–.0045 in.
Piston pin to conn. rod bushing clearance (max. allow. worn)	.015 in.
PISTON AND LINER	
Liner diameter (new)	8.4995–8.5005 in.
	8.449 –8.451 in.[3]
Piston diameter (new)	8.474 –8.476 in.[4]
	8.486 –8.488 in.[5]
Piston to liner clearance (new)	.0485–.0515 in.[6]
	.0235–.0265 in.[7]
	.0115–.0145 in.[8]
Piston to liner clearance (max. allow. worn)	.075 in.[6]
	.050 in.[7]
	.040 in.[8]
Piston diameter (min. allow. worn)	8.475 in.[8]
Liner diameter (max. allow. worn)	8.525 in.
Liner out-of-roundness (max. allow. worn)	.007 in.
Piston pin bushing inner diameter (new)	3.0005–3.0015 in.
Piston pin bushing inner diameter (max. allow. worn)	3.008 in.
Piston pin to piston bushing clearance (new)	.0005–.0025 in.
Piston pin to piston bushing clearance (max. allow. worn)	.016 in.

TABLE 7-1.—(Continued)

PISTON RINGS	
Compression ring gap clearance (new)	.030 –.050 in.
Compression ring gap clearance (max. allow. worn)	.100 in.
Oil control ring gap clearance (new)	.030 –.050 in.
Oil control ring gap clearance (max. allow. worn)	.100 in.
Compression ring side clearance (new)	.008 –.0105 in.[9] .004 –.0065 in.[10]
Compression ring side clearance (max. allow. worn)	.020 in.
Oil control ring side clearance (new)	.002 –.0045 in.
Oil control ring side clearance (max. allow. worn)	.020 in.
CAMSHAFT	
Shell thickness (new)	.1222–.1225 in.
Shell thickness (min. allow. worn)	.117 in.
Shell to shaft clearance (new)	.0035–.0061 in.
Shell to shaft clearance (max. allow. worn)	.010 in.
VALVES	
Exhaust valve tappet clearance (cold)	.015 in.
Exhaust valve guide diameter (new)	.565 –.566 in.
Exhaust valve guide diameter (max. allow. worn)	.573 in.
Exhaust valve guide to valve stem clearance (new)	.002 –.004 in.
Exhaust valve guide to valve stem clearance (max. allow. worn)	.012 in.
MISCELLANEOUS	
Injector nozzle opening pressure	1000#[11] 3200#[12]
Injector timing (BTC)	7°[13]
Lube oil recommended	9370 9250[14]
Cylinder head stud torque lb-ft	650
Cylinder liner stud torque lb-ft	250
Conn. rod bolt torque, lb-ft	100

[1] For Satco shells only—renew Trimetal shells when intermediate bronze lining starts to show through.
[2] Renew when backing starts to show.
[3] Top of taper, at top of head.
[4] Bottom of layer, top of 5th ring groove.
[5] Skirt from 5th ring groove to bottom.
[6] At top of piston.
[7] Between 4th and 5th ring grooves.
[8] At skirt.
[9] First and second rings from top.
[10] Third, fourth, and fifth rings from top.
[11] Spherical check-valve type.
[12] Needle-valve type.
[13] Position of flywheel for checking injector with injector timing tool
[14] First choice substitute.

equipment in question. Makers of the auxiliary equipment, generators, switchboards, cooling systems, air filters, and the like issue their *own complete instruction books.* All of this information should be accumulated in one place, so that it will be easily accessible to the operator. Such information would make another complete textbook—a brief résumé only can be given here.

Switchboard. Clean with air, thoroughly removing accumulations of dust and deposit of all kinds.

1. Test for grounds and check ground connections.
2. Inspect all control instruments and contactors.
3. Test watt-hour meters and the like.
4. Check, adjust, and make record of relay settings.
5. Clean and inspect all bushings, connectors, and switches.

Generators. Clean the windings with air, and paint if that is necessary. Check lubrication and clean bearings when needed.

1. Inspect collector rings and brushes.
2. Check the air gap on all four sides.
3. Test frame grounds and make insulation test.

Exciters and small motors. Clean with air and paint the windings when this is indicated.

1. Check the air gap and bearing alignment.
2. Check couplings, pinion, or pulley alignments.
3. Inspection of the electrical features such as the commutator brushes and the making of an insulation test should be done by a competent electrician unless the operator is experienced.
4. All auxiliary equipment motors should receive periodic check and inspection.

Electrical starting equipment. Clean with air, inspect, and oil. Check, adjust, and make a record of undervoltage and overload relays setting. Inspect and test switch, conduit, and motor frame grounds. Inspect all electrical equipment in the engine room.

Jacket water heat exchangers. Inspect regularly for leaks and for scale. Clean the tubes before the deposits reach an appreciable thickness. The kind of scale should also be determined, and the method of preventing this formation worked out if possible.

Cooling system pumps. Check pumping level, flow, power input to the pumps, and the speed of the pumps. Drain and renew oil in the pump bearings. Check thrust bearings and clearance, and at intervals, pull and inspect the pumping unit.

1. Check suction pressure.
2. Check discharge pressure and flow.
3. After careful internal inspection, drain and renew the oil.

Air compressor. Drain and renew the oil at proper intervals, and overhaul when so indicated. Inspect valves and bearings as previously outlined. Give the compressor a major overhaul when the general condition requires it.

Fuel-oil-storage tanks. Drain and clean out periodically, and inspect for corrosion. Drain off the water regularly and inspect for leaks. The outside of the tank should be cleaned and painted to prevent any rust accumulations.

Air-storage tanks. All air-storage tanks should be drained regularly and should have the hydrostatic test applied for safety purposes. Water accumulating from condensation is the source of trouble and must not be neglected for long intervals.

Fire-safety equipment. Inspect and list all extinguishers. Discharge and recharge and soda acid and foam type; test and fill the tetrachloride type. Obtain the Underwriters' Regulations for Internal Combustion Engine Plants and consult the National Safety Council for additional information on safety in such plants.

Water and oil piping. Inspect all piping for leaks; clean and keep painted all exposed pipe; do not permit formation of rust on the exterior or corrosive scale on the interior of any piping. The oil filters and strainers, supply lines, heaters and auxiliary storage tanks should be inspected and cleaned regularly.

Air intake and exhaust equipment. Inspect and service the air filters regularly; keep the air suction ducts and mufflers cleaned. This also applies to exhaust ducts and mufflers. Inspect the gas flow regulators if natural gas is used for fuel.

Relief valves. All relief valves, for air-starting systems, lubricating oil pressure control, and so on, should have careful attention. The safety or relief valves on the cylinder heads must not be neglected.

The auxiliary and accessory equipment manufacturers willingly furnish completely reliable information on the operation, maintenance, and installation of the apparatus furnished the engine manufacturer. It is the duty of the operator and maintenance man to make sure he has a complete set of such instructions covering all items of auxiliary and accessory apparatus. A careful study of this class of information is recommended.

It is obvious that a log book should be kept on each major unit of auxiliary equipment and all maintenance and operating data entered in this log.

QUESTIONS

1. What are the important factors in maintenance and details of keeping accurate records?

2. What clearances and adjustments must be made and kept at all times?

3. Why is it important to maintain the proper compression pressure at all times?

4. Why should the rocker arm clearances be kept absolutely standard?

5. How are the clearances maintained in the valve mechanism?

6. How is valve timing checked?

7. What condition of the pistons and liners should be maintained with respect to clearances?

8. What is the correct guide for piston end clearance?

9. What causes piston end clearance to vary?

10. How may piston end clearance be maintained?

11. What wearing parts require the most attention?

12. What safety precautions should be taken in the operation and maintenance of the engine?

13. What steps should be taken to avoid dangerous fires?

14. How is valve timing checked?

15. Describe the procedure for valve timing.

16. Why is it necessary to reset valves and when is this to be done?

17. What is the procedure for checking and inspecting the air-starting system, the fuel system, and the lubricating system auxiliary equipment?

18. What information on the auxiliary equipment should be obtained and kept conveniently for the operator; where is it obtained?

CHAPTER 8

PISTON RING MAINTENANCE AND INSPECTION

Nomenclature. Piston and ring nomenclature has been standardized to a great extent, particularly for the automotive field. The Gasoline Engine and Diesel Engine Divisions of the Society of Automotive Engineers submitted final recommendations for approval and publication in the 1944 *SAE Handbook*. The project was suggested by Mac O. Teetor in the *SAE Journal*, September, 1944. The original suggestion is shown in Fig. 8-1 and Fig. 8-2 herewith. These terms are generally used with some modification by all writers.

Fundamental function of piston rings. It may be said that the piston and the cylinder are a pair of elements, one working in the other, the piston sealing being done by rings. The difference in diameter of the working piston and the fixed cylinder is sufficient to allow for free movement under operating conditions. The function of the rings is to prevent leakage between the piston and the cylinder when operating under pressures of compression and combustion.

Piston rings must have flexibility. In order to accommodate itself to the irregularities and uneven wear of the liner, the ring must have a certain "spring" or flexibility, and also be able to exert an adequate pressure against the cylinder wall and against the land of the groove. The tension of the ring and its flexibility are two characteristics very closely related.

Tension or pressure. The tension or spring of the ring must be predetermined. It ranges from 5 to 8 psi and sometimes higher. The ring should have only the tension necessary to perform its work. Too much tension makes it difficult to spring the ring apart to slip it over the piston without breakage or distortion. This tension is built into the ring by the manu-

facturers. Whatever the method used to impart tension to the ring, it must exert a uniform pressure around the cylinder.

The method used to impart tension to the ring is hammering, rolling, and casting in an out-of-round form. The purpose of

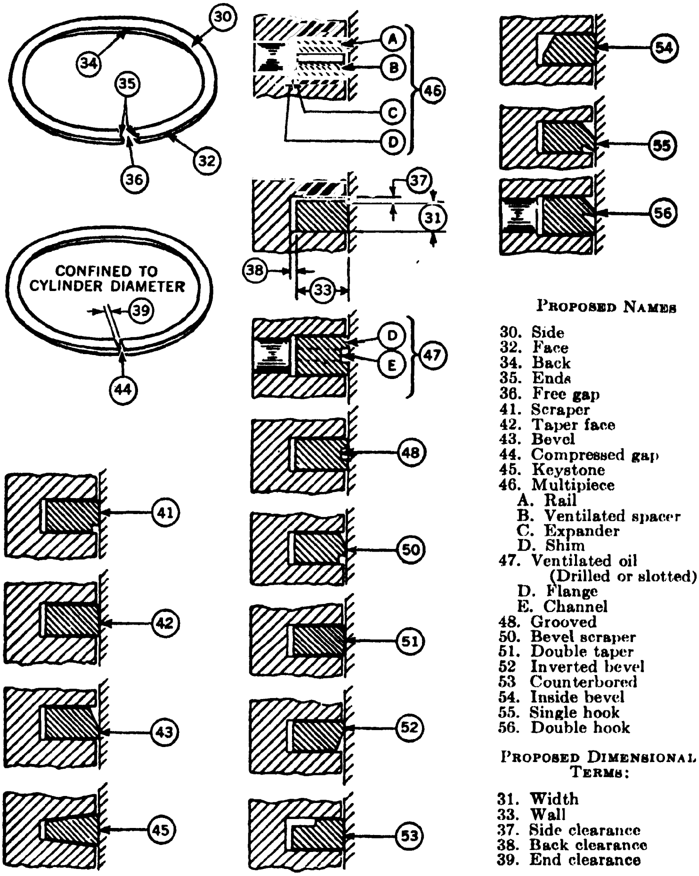
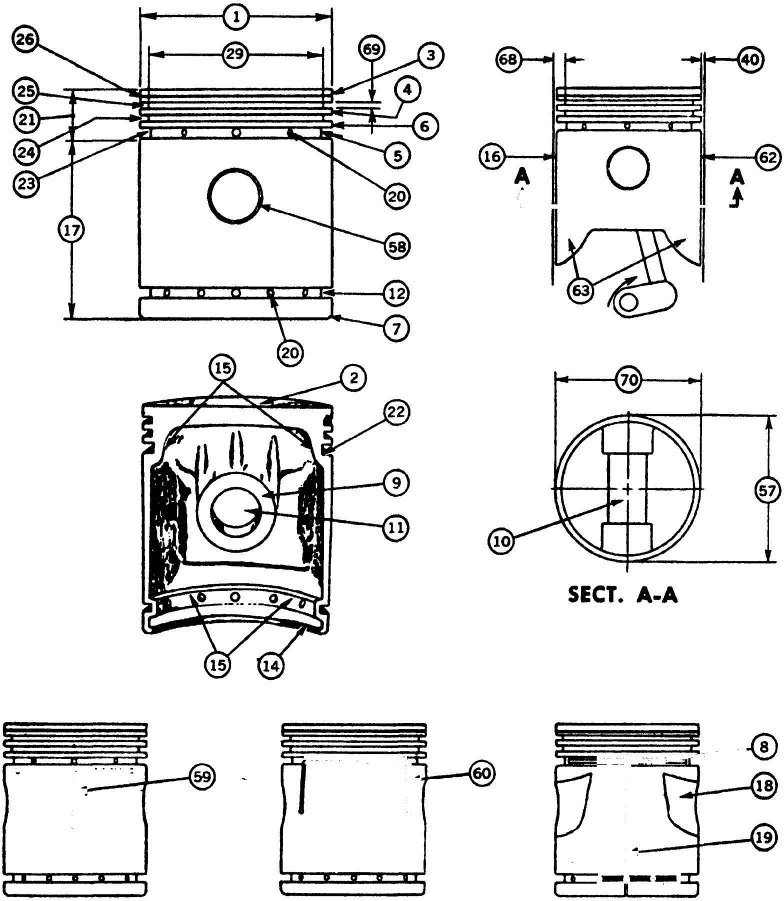


FIG. 8-1. Piston ring nomenclature recommended by the SAE General Standards Committee and published in the *SAE Handbook.*

any method is to control uniformly the pressure that the ring exerts from point to point around its circumference. When a ring does not exert uniform bearing pressure even during the wearing-in period, brown or burned spots will be found on the rings where the pressure is too low and blow-by occurs.



PROPOSED NAMES:

2. Head
3. Top land
4. 2nd land
5. Ring groove bottom
6. 3rd land
7. Chamfer
8. Horizontal slot
9. Pin boss
10. Piston pin
11. Pin hole
12. Skirt ring groove
14. Bottom rib
15. Ring groove pad
16. Major thrust face
17. Skirt
18. Skirt relief
19. Vertical slot
20. Oil drain holes
21. Ring belt
22. Ring groove side
23. Oil ring groove
24. 2nd ring groove
25. Top ring groove
26. Heat dam
58. Pin bushing
59. T-slot
60. U-slot
62. Minor thrust face
63. Slippers

PROPOSED DIMENSIONAL TERMS:

1. Land diameter
29. Groove bottom diameter
40. Land clearance
57. Minor diameter
68. Groove depth
69. Groove width
70. Major diameter

FIG. 8-2. Automotive engine piston nomenclature recommended by the SAE General Standards Committee and published in the *SAE Handbook*.

Piston ring friction. Rings operate with considerable friction, even with proper lubrication. The tension exerted by the ring, plus the pressure of the gas behind the ring, forces it against the cylinder wall with considerable pressure, usually close to that existing in the cylinder above the piston. This pressure caused by the gas is greatest for about 15 per cent of the piston travel measuring from the firing stroke. The liner at the lower end of the piston stroke will have little measurable wear, which shows that the pressure of the ring against the cylinder liner is very little at the end of the ring travel.

Sealing action of rings. The direct-sealing action of the ring by its bearing against the cylinder liner as a result of its tension alone would be insufficient for sealing the cylinder. The use of more than one ring, each ring in its turn partially holding the pressure leaking from the ring above, steps down the pressure to a low value, the number of rings depending on the type and design of engine. This is referred to as the "labyrinth principle," the rings providing a tortuous path presenting resistance to the escaping pressure by forcing it out of a direct path into a labyrinth path.

The gas pressure holds the ring down against the lower side of the ring groove, on the land, and at the same time, forces it out against the cylinder wall. The ring gaps are staggered—an arrangement that provides the restricted path in the space above and behind the ring. The gas pressure can pass from one ring gap to another only by transversing 180 deg of the ring circumference, back and forth. Thus the pressure continues to flow from ring gap to ring gap, each step reducing its velocity and therefore lowering its pressure.

Number of rings per piston. When a sufficient number of rings are used, the pressure of the gas leaking past the piston and the last ring is negligible, and the amount of gas small, but even if the initial amount of leakage is considerable, the pressure is harmless. This assumes that the side and back clearances are kept at the right value and that the end gap or free gaps are not too great. When rings have proper fit, there is no harmful blow-up until the liner is worn beyond the permissible limit and allows too much free gap clearance.

Seating of rings. A high polish on the wearing surface and the proper seating of the ring against the cylinder wall and on the land of the groove are extremely important. When the ring has seated properly, it has acquired a high polish. Then

only a small amount of gas can get between the ring and the cylinder wall to exert pressure tending to separate them. There is much greater pressure above the ring and behind it in the back clearance than between the ring and the cylinder liner wall, so that the ring will be held outward at all times unless there is the condition of *ring flutter*.

Side clearance. This is the difference between the width of the ring groove and the thickness of the ring. This must be kept at the minimum consistent with proper freedom of the ring in the groove. It is the purpose of maintenance and inspection to keep it thus. The size of the piston and the position of the ring on the piston as well as other factors determine the side clearance, which may range from 0.002 to 0.007 in. When grooves are worn, or there is too much clearance, carbon accumulates in the back clearance, and between the ring and the top of the groove to a point where sticking can occur.

Shouldering of the ring grooves. The enlargement of the ring groove width occurs after a long period of operation. It is said that the ring actually hammers out the metal of the ring groove land on account of the reciprocating motion of the piston. Once excessive side clearance exists as a result of gradual wear of the ring, the hammering-out action of the ring on the lower ring land increases rapidly; hence the need for replacing worn rings when the wear has been sufficient as determined by experience. The rings hammer out a shoulder on the lower ring land owing to the fact that the radial depth of the ring is less than the radial depth of the groove to permit the movement of the ring, so that as the ring wears, the shoulder is formed. Such a shoulder interferes with the proper fitting of a new ring. The shoulder must be removed by machining-out of the groove to remove the shoulder so that an overwidth ring can be fitted.

New rings should never be fitted in shouldered grooves. The ring wears away a width of metal exactly its own width while some metal is left standing near the bottom of the groove in the back clearance. When a new ring is fitted, its radial depth is greater than the worn ring removed, and thus the new ring will foul and stick on the shoulder left standing in the groove. This is the reason shoulders must be removed before new rings are fitted.

Failure of rings to seat. If rings do not seat properly or for any reason there is serious blow-by, the hot gases flowing

between the piston and the cylinder wall will burn off the lubricating oil. When metal to metal contact follows, pistons seize and generally there is mechanical breakdown of the engine in short order. It is important that rings be properly fitted and seat adequately in a short time.

Testing for leakage. The compressed-air test is used in air-starting engines to check on the tightness of the rings. The engine is placed on top dead center for one piston and the

INSPECTION OF RING AND RING LANDS Cylinder No.______

DEPOSIT		RING GROOVE																											
Quadrant		0° - 90°							90° - 180°							180° - 270°							270° - 360°						
RING GROOVE		1	2	3	4	5	6	7	1	2	3	4	5	6	7	1	2	3	4	5	6	7	1	2	3	4	5	6	7
Character	Symbol																												
Light Lacquer	LL						✓	✓						✓															
Soft Sludge	SS					✓		O						O								✓							✓
Soft Carbon	SC			O	✓								✓								O	O						✓	O
Med. H Carbon	MH	O		✓								✓	O						O	O ✓	✓						✓	O	
Lacquer	L		✓						O	O	O ✓	O						O ✓	✓				✓	✓	✓	✓	O		
Hard Carbon	HC	✓							✓	✓						✓	✓						✓	O		O			
Very Thin	VT	✓							✓	✓						✓	✓							✓					
Thin	T		✓	✓	✓	✓					✓	✓						✓	✓						✓	✓	✓		
Med. Thick	MT						✓	✓					✓	✓	✓					✓	✓							✓	
Thick	TT							✓														✓							✓

Check mark (✓) condition of rings

Symbol (O) condition of ring land.

FIG. 8-3. Form for inspection of rings and ring lands deposits (ASME *Mechanical Engineering*). (A form like this report form should be used.)

starting air turns on, or air pressure is turned on the cylinder on the firing position. With an air-pressure gauge placed in the indicator cocks, the fall-in pressure, after the air is turned off, indicates the rapidity of leakage. If the rings are stuck, or not seating sufficiently to hold the pressure, the pressure will blow down rapidly when the starting air is cut off. The escape of air can be heard at the crankcase openings.

These tests should be made immediately upon shutting down the engine since a cold engine may blow down quickly when in good condition. When very cold, the lubricant is sticky in the ring grooves, the rings are sluggish, and leakage is rapid.

Removing stuck rings. Taking the stuck rings off the piston is made easier by soaking the piston in kerosene, or by heating and boiling them for some time in soapy water. When the rings are burned in place and stuck with hard carbon and these methods do not loosen them, the rings must be broken out. A brass probe is used, together with a hammer. Using a chisel is not recommended, for this method damages ring grooves.

Inspecting and installing rings. Unless the ring is perfectly flat, it may bind in the groove. Rings can be warped by handling or dropping. It is good practice before installing rings to place them on a surface plate or a piece of plate glass. One test is to lay pieces of cigarette paper under the ring around the circumference at several points. The delicate hand can determine how much tension is exerted to pull the cigarette papers out. This method helps to determine whether the rings are bowed and fail to lay flat.

The best procedure when installing rings is to go over the ring with a smooth file to remove the burrs and dirt from the edges of the ring, particularly removing the burrs from the inner edges and gap ends. The edges of the ring grooves should also be free and smooth; a used file can be applied to smooth them. Any dents or burrs should be removed from the ring lands on the piston.

The rings should be cleanly wiped, with clean cloth, not with waste. Removal of dirt and foreign matter is important because small particles of dirt and grit will fill up the effective clearance and impair the freedom of the ring to move in its groove.

Distortion of rings. It is necessary to avoid distorting the ring when it is being placed on the piston. The use of metal strips is practical for this purpose. Some operators use a ring pot for this work.

Back clearance. The difference between the radial depth of the ring and that of its groove is the back clearance. This must be sufficient at all times to clear the ring when pushed down into the groove and enough to prevent the ring bearing hard against the liner, even with some carbon formed behind this ring. If the ring projects beyond the piston when bottomed in the groove, the side thrust of the piston causes rapid wear of ring and liner. It is possible that much cylinder wear is traceable to this condition, which occurs when the ring grooves are permitted to accumulate carbon in the back clearance space.

This formation of carbon should be cleaned out by pulling pistons as often as necessary.

Gap clearance. Rings elongate or expand when warmed up in service, and the ends of butts will touch unless the gap or end clearance is kept sufficient to allow for this. The rings, when they touch ends, form a solid, rigid arch around the cylinder and break quickly, with scoring of the cylinder by partial seizing. Whenever rings are broken in operation, it is likely that the ends of the butts will show bright spots indicating lack of clearance. Proper gap clearance depends upon the size of the engine and other factors. The rings should be fitted for end gap clearance at the smallest section of the cylinder, usually near the bottom of the ring travel. When the gap clearance is too large, there is a definite increase in blow-by.

Cyl. No._____		RING STICKING							
Ring No.		Symb.	Compression					Oil	
Cyl. Condition			1	2	3	4	5	1	2
Free							✓		✓
Sluggish		SS				✓		✓	
Pinched	0° - 90°	LL			✓				
	90° - 180°	LL		✓					
	180° - 270°								
	270° - 360°								
Stuck	0° - 90°								
	90° - 180°	MH	✓						
	180° - 270°								
	270° - 360°								

FIG. 8-4. Report form for ring sticking inspection.

Methods of reducing gap leakage. Since a large part of the total amount of gas escaping the rings passes through the end gap, efforts to limit this have been made by the use of bevel or angle-cut, or step-cut design. A beveled ring may be fitted closer than a square-cut joint or a step-cut joint since the difference in dimensions of the gap is measured at right angles to it, and along the line of the piston, and the clearance may be less than for other joint types. This difference is very small, however, and therefore straight-cut or square-cut joints are often used for the sake of simplicity in manufacturing and servicing.

Other methods to reduce gap leakage are the use of the two-piece and three-piece rings, especially in worn cylinders, such as the *Double Seal* ring and many types similar to it. Installation of any of these special rings requires careful attention to the instructions for using them. Good results in worn cylinders are obtained with such rings when properly installed.

Inspection of rings and ring grooves. Under actual service conditions, many problems of operation and maintenance are traced to the piston ring and ring groove conditions. Several

factors are related to ring troubles, or the result of ring troubles:

1. Deposits, foreign matter, and abrasives.
2. Faulty, incomplete lubrication and the use of poor lubricating oil unsuited to the engine design.
3. Abuse of engine, overload, and so on.
4. Blow-by caused by worn rings and grooves.

When pistons are pulled for inspection and overhaul, a careful examination of the rings is made, and a study of the data and their relation to the factors involved in ring troubles reveals causes that can be eliminated. There are sufficient and good

AREA ABOVE TOP RING - CONDITION Piston No.____					
QUADRANT	0°-90°	90°-180°	180°-270°	270°-360°	Area
Character	T - Thin	MT - Medium	TT - Thick		
Soft Carbon	T			TT	
Med. Carbon	✓	MT		X	
Lacquer			T	T	
Hard Carbon	T	X	O		

(✓) Scuffed light.
(O) Scuffed deep.
(X) Scuffed scored.

Remarks:

Disposition:

FIG. 8-5. Report form for deposits above top ring.

reasons for making these careful inspections and attempting to evaluate the data observed, which will become more apparent as the inspection progresses. The object of such inspection is to find the basic reasons for the troubles. The inspector looks for the following evidence when the rings are inspected:

1. Lack of lubrication.
2. Corrosion of engine parts and rings.
3. Ring sticking and clogging of oil channels.
4. Excessive ring wear, including breakage.
5. Excessive carbon deposits, lacquer formation, and deposits in ring grooves.

The piston and rings are carefully examined, and observations are made for the following items, which are steps in the inspection procedure:

1. Ring groove deposits.
2. Piston land deposits.

3. Amount of deposits above the top ring.
4. Deposits in the oil ring slots and holes.
5. Sludge deposits in the lubricating oil system.
6. Sludge and lacquer on the piston, liner, and so on.
7. Scuffing and scoring of the liner.
8. Cylinder liner wear.
9. Top ring wear.
10. Corrosion of parts.

Suitable forms for making records of the inspection are suggested. Some forms are given here to indicate the extent of the attention that may be given this phase of engine maintenance. These have been used for recording endurance test data on lubricating oil, rings, and the like.

Ring sticking. When pistons are pulled for ring inspection, a careful procedure should be followed in making visual observations and a study of the condition of the engine, and the rings. The performance of the lubricating system, the operation of the engine in general, and the question of a top overhaul are involved. Engine cleanliness is the desirable condition. The engine is either clean and the piston and rings are operating satisfactorily without undue wear, or it is at some stage between the clean condition and the worst condition that could be expected. The necessary observations to be made and the method of indicating these observations on a report form widely used in engine testing and operation is indicated in the illustrations of the report. A glance at this report shows:

1. That one of these rings is entirely free in the grooves.

2. That one ring is stuck on the second quadrant, two rings are pinched, one sluggish, and one may be considered free.

3. The deposits range from *medium-hard carbon* on the ring land to *hard carbon, very thin* on each of the four sides or quadrants. This is ring No. 1. As other rings are inspected, the deposits on the rings and ring grooves range from hard carbon to light lacquer for the sixth ring. The condition of the area above the top ring on the piston is also indicated as having deposits of medium carbon, hard carbon, and very thick, soft carbon. Moreover, it is scuffed in some places and has deep scores in other places.

Degree of ring sticking. It may be noticed that the rings are free, sluggish, pinched, or stuck. The location of the condition as well as the nature of the deposit is also indicated. The

explanations of the various conditions are given in the following outline:

a. A *free* ring is one that falls in its groove of its own weight when the piston is moved from the vertical to a horizontal. Unless it does so, it is:

b. *Sluggish,* in which case it will not fall of its own weight when the piston is moved from a vertical to a horizontal position, but is easy to move with moderate finger pressure. If it is not, it is:

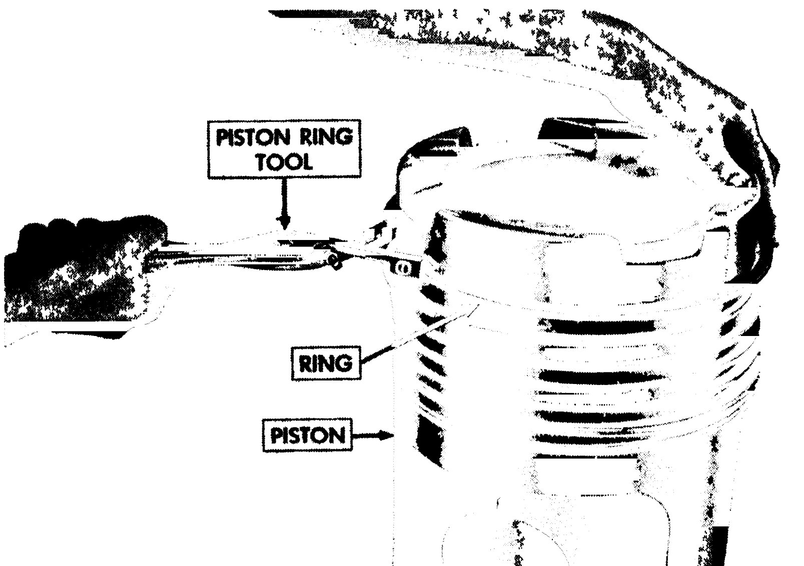


Fig. 8-6. Installation of piston rings, showing use of piston ring tool.

c. *Pinched.* In this condition it does not move in its groove under moderate finger pressure, but it will have a bright or well-polished face, an indication that it is essentially still free during operation, but that deposits are at a maximum in the groove.

d. A *stuck* ring is one that does not move under moderate pressure, and, at the same time, its face is covered with lacquer, or carbon deposits of various thicknesses appear over any or all parts of its circumference. It may be partially or completely stuck, as a result of these deposits, warped conditions, and so on. A stuck ring does not move in its groove and does not bear against the cylinder wall during the operation of the engine.

Deposits above the top ring. The deposit above the top ring is very closely related to the combustion efficiency and the tendency of the lubricating oil to form carbon. The deposit may be slight and unimportant, or it may be in any condition ranging up to a point where the area above the top ring is covered 100 per cent with thick, hard carbon or lacquer completely fills the clearance space between the piston and cylinder. The percentage of the area covered by these deposits as well as the nature of the deposits themselves is noted. The thickness of the deposits is also observed. This may range from a *thin* coating taking up about one fourth of the clearance between the piston and cylinder to a *medium-thick* coating, about three fourths of the clearance, or to a *thick* coating, completely filling the clearance space. The characteristics of the deposits should also be noted. These will range from soft carbon, medium-hard carbon, lacquer, to hard (brittle) carbon.

Ring groove deposits. The circumferential area of the inner wall of the groove may be covered 100 per cent with thick, hard carbon or lacquer, or it may be clean and free of deposits. The thickness of the deposits should be determined at four positions on the inner wall of the groove in the same manner as indicated for the rings. The percentage of the area covered should be noted. Again here the thickness of the deposits should be noted and will be (1) clean, (2) very thin, (3) thin, (4) medium-thick, or (5) thick. The character of the deposit will be (1) light lacquer, (2) soft, oily sludge, (3) soft carbon, (4) medium-hard (flaky) carbon, or (5) hard (brittle) carbon.

Piston land deposits. The area of the lands between the compression rings is either clean or covered with thick, hard carbon, lacquer, medium carbon in a condition that may be described as very thin, thin, medium-thick, or thick. These conditions have the same meaning as for the area above the top ring as to relation of the thickness of the deposit to the clearance between the piston and the cylinder.

Compression ring wear. It is considered important to determine during an inspection of the cylinder and piston the amount and rate of wear of the compression rings. The top ring is weighed and the percentage of loss in weight computed on the basis of 1000 hr of engine operation. For small engines of less than 300 hp, this should be less than 10 per cent loss in weight over its original weight. A number of top rings should be weighed and a number of new rings weighed and the loss

computed on the basis of the 1000-hr period. For larger engines, another method may be used. The maximum rate of wear should be when the percentage of assembled butt-clearance increase of the compression ring in the top groove becomes 300 per cent of its original value, computed on the basis of 1000 hr of engine operation. Excluding the initial wear-in operation period, the first check should be made after not more than 450 hr of actual operation. When it is desired to check the wear very

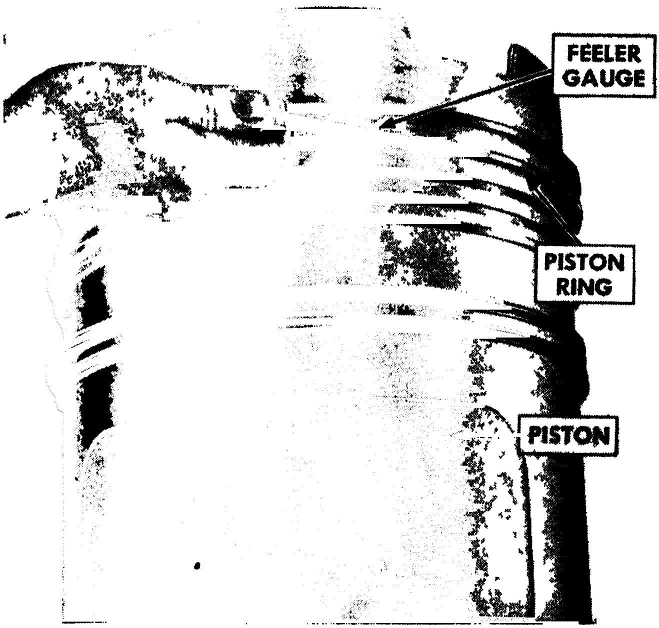


FIG. 8-7. Use of feeler gauge to measure side clearance.

carefully and accurately during the first few months of operation of a new engine, the check can be repeated every 250 hr or oftener. When a new ring design test or a test for the merits of a lubricating oil is desired, careful measurement of ring wear is important.

Scuffing above the top ring. The worst condition usually found is that the entire area above the top ring is deeply scuffed. Engines that have any of this area scuffed to any considerable extent are far from operating satisfactorily. The scuffing may be "light" or "deep" and should be so indicated and shown on the report. When this part of the piston is scuffed or scored

deeply, the engine piston is running hot and may be distorted or expanded, as a result of heat, to such an extent that it must be replaced to avoid continued damage to the cylinder liner.

Analysis of inspection reports. It is obvious that several uses can be made of the data that result from such a step-by-step inspection of the piston and rings. The following are among the obvious things that can be done:

1. The period between inspections and overhaul can be more accurately determined.
2. The rate of wear and the need of replacement of rings can be determined.
3. The suitability of the lubricating oil, the fuel oil, and the engine application would be indicated over a period of time.
4. Under some circumstances, it would be advisable to make such information available to the engine builder, who can advise steps to be taken in the event of a definitely unsatisfactory operation.
5. These observations of conditions of the engine piston and rings have a definite relation to the kind of lubricating oil used and the service it is giving. The supplier of the oil would immediately advise concerning a change in specifications in the event that the operation was not satisfactory or the oil was not giving good performance.

It is evident that careful inspection and use of the resulting data contribute importantly to satisfactory and economical

TABLE 8-1
PISTON RINGS—GAP AND SIDE CLEARANCES
(GENERAL MOTORS MARINE ENGINE)

Clearance	Amount of Clearance
1. Oil control—gap clearance (new).....	0.030–0.050 in.
Oil control—gap clearance (worn)....	0.100 in. (maximum)
2. Compression—gap clearance (new)....	0.030–0.050 in.
Compression—gap clearance (worn)...	0.100 in. (maximum)
3. Compression—side clearance (new)...	0.008–0.0105 in. (1st and 2nd rings from top)
Compression—side clearance (worn)..	0.020 in.
4. Oil control—side clearance (new)......	0.002–0.0045 in.
Oil control—side clearance (worn).....	0.020 in. (maximum)
5. Compression—side clearance (new)....	0.004–0.0065 in. (3rd, 4th, 5th rings from top)

NOTE: These clearances are typical of a number of engines of this size and speed (a 9.50-in. piston, operating at 750 rpm).

operation. Any maintenance man or engine operator can accumulate such data with very little additional effort when provided with the necessary instructions and forms for recording the same. All piston ring gap and side clearances should be checked each time the rings are inspected.

It should be specially observed that the side clearance is due both to the wear on the rings and ring groove lands. It is essential to know what part of the clearance is due to groove wear and what amount is due to ring wear. It is obvious that when the ring grooves have been worn 50 per cent of the maximum allowable clearance for a worn ring, the new ring installed will give only half the service that it would in a new ring groove. As the ring grooves wear, the life expectancy of the rings becomes less.

TABLE 8-2
PISTON TO LINER CLEARANCE
(GENERAL MOTORS MARINE ENGINE)

Clearance (new)	Amount of Clearance
At top of piston	0.079–0.0825 in.
Between 4th and 5th ring grooves	0.0505–0.0525 in.
Junction of taper below 5th ring	0.0385–0.0405 in.
Bottom of lower taper at lower end	0.0315–0.0335 in.

From the data in the above table, it can be seen that checking piston to cylinder clearance against the thickness of the deposit on the piston above the top ring and the piston land and the ring groove should be accurately determined. Once this clearance space fills up with carbon or sluggish lacquer, the piston drags and fails to dissipate heat, and the condition, if it continues, contributes to ring sticking. Carbon deposits on these areas function as heat insulators, preventing adequate cooling of the piston and rings. It has been said that the section of the piston carrying the rings dissipates around 50 to 60 per cent of the heat that must leave this section of the piston. The balance of the heat absorbed by this part of the piston must be radiated to the cooling jackets by transfer direct from the piston to the liner wall, and to some extent, to the lubricating oil. Reducing the capacity of the piston to dissipate heat endangers the operation. It is therefore obvious that keeping the piston comparatively free of deposits better insures satisfactory

operation by permitting cooler operation and balanced heat disposal.

Many small engines have top rings installed with side clearance of 0.002 to 0.003 in. with pistons between 3 and 5 in. in diameter. A minimum rule is a clearance of 0.004 in. per inch of piston diameter (and for some of the older, large slow-speed, heavy-duty engines, as much as 0.006 in. per inch of diameter) for the top ring of large engines. This would be from 0.040 to 0.060 in. for a 10-in. diameter piston.

A difference in temperature of 200 to 300° F between the piston and the cylinder wall is not uncommon in Diesel engine operation. Since the rings will attain about the same temperature as the piston, especially the top ring, and since the temperature of the cylinder wall is not less than that of the cooling water, the ring temperature is certainly around 320° F. Should this temperature increase as a result of insulating deposits and encrustations in the grooves, on the lands, and above the top ring, and reach a value of 400° F, most lubricating oil would begin to break down and form carbon. The heat and the temperature effects caused by piston deposits are progressive; a slight increase in the thickness of the deposits will give an increasingly large rise in piston and ring temperature. The temperature soon reaches the point where the lubricating oil is baked, forming gummy deposits of lacquer and eventually hard carbon, sticking the rings, and contributing to a condition of unsatisfactory operation.

Cleaning carbon deposits. In the early days of the Diesel engine, cleaning the carbon deposits was a frequent occurrence. Modern design has reduced this to a low value, but the engine must be properly operated and preventive maintenance observed, for removal of deposits is one of the chief reasons for a schedule of inspection and preventive maintenance.

Vibration or ring flutter. In some cases, it has been reported that ring failure and ring troubles are due to vibration or flutter. The rings are forced by contributory vibration to reach a periodic vibration. When a ring attains this periodic vibration, it is said to remain in constant radial movement between the cylinder wall and the bottom of the groove. This permits explosion pressures to pass the rings, and also permits oil to get into the combustion chamber. Well-fitted rings with minimum clearance are less likely to behave in this manner. It is doubtful if the condition described is very common,

but it is a matter for consideration if breakage occurs without an obvious cause being evident. Any absolute evidence of this kind would be confined to the high-speed Diesel engine. Too little gap clearance is usually the cause of breakage of rings in operation. When these engines are started, the rings absorb heat more quickly than the cylinder liner expands; the ends

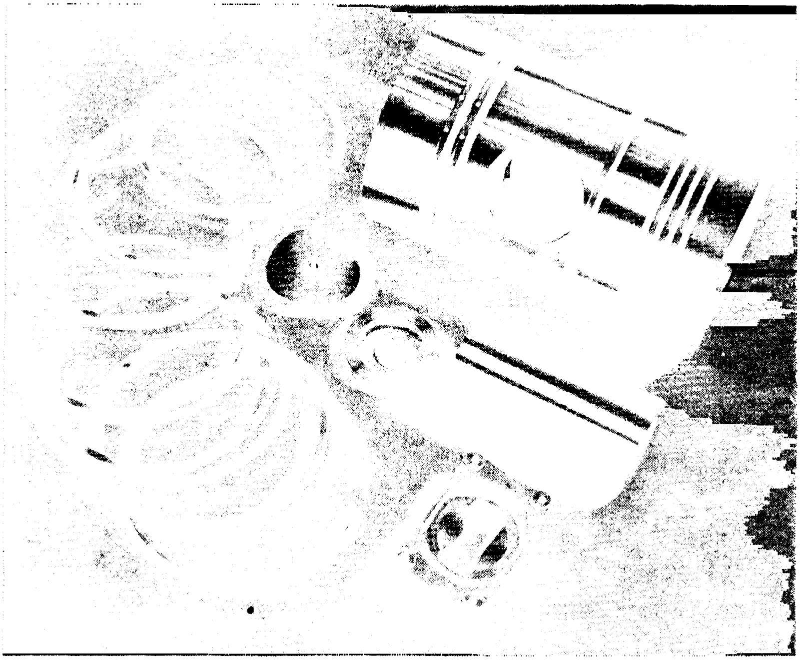


FIG. 8-8. Complete piston assembly (Hendy Series 50 engine). Oil-cooled pistons improve ring and cylinder life, so that rings run cooler. Heavy ribs of the underside of the piston crown promote rapid heat transfer to the oil. The cooling chamber in the piston head is designed to remain half full of oil under all conditions of operation.

butt and break. It is true that the inertia forces of the ring mass tend to throw the rings away from the groove during the latter part of the stroke and the first part of the returning stroke. This is negligible in engines with low piston speed and equipped with narrow rings, and it is likely that the rings remain in the position required during the compression and expansion stroke. However, if the rings do leave their seats near the end of the piston stroke, owing to the inertia of the heavy rings, there

will be some gas leakage between the rings and the groove seats. This would usually be mistaken for blow-by through the gaps, especially when it is sufficient to cause a loss of power. A cylinder and piston, with grooves in good condition and with rings in intimate contact with the cylinder wall, should have very little blow-by past the face of the rings if these rings do not vibrate in and out of the grooves. Blow-by between the butts of the rings is comparatively small and insignificant in its effect upon the lubricating oil film as compared with the leakage between the face of the rings and cylinder wall, if indeed this should occur. The matter is worthy of investigation when there is evidence of its occurrence.

Special ring designs. Many special designs of piston rings are offered as a solution to the problem of holding the compression in Diesel engines, especially with worn cylinders. Such rings are usually wider than plain rings, and special groove width should be provided to use them. These rings also cost more than plain rings, but in many cases, the cost is insignificant in comparison with the additional life that may be obtained from worn liners and pistons when such rings as the one-piece *Double Seal* type are used. The gap clearance is no longer a question with these gap-sealing designs—the feature which justifies their use in worn cylinders. They are designed to seal the gap as well as the groove of the piston. The installation of such rings is more important than that of plain rings. They must be installed in the proper grooves and should not be placed upside down. It must also be determined whether they can be used in the top groove and how many are needed in addition to plain rings, since some plain rings can usually be used in conjunction with the sealing-type ring.

Installation procedure. The installation of piston rings with a lack of sufficient side clearance in the top groove may result in the rings sticking and, very often, breaking in a very short time. Another factor that can cause breakage of piston rings is the installation of the rings with a lack of sufficient end clearance. Piston ring lands are easily fractured by removing the piston from the cylinder without removing the *ridge* at the top of the cylinder. When the engine cylinder has been badly worn, leaving a considerable ridge at the top of the cylinder, and the piston is pushed past this ridge, the breakage or fracture of the ring land between the first and second rings may occur.

In order to avoid ring and ring groove land breakage from these sources a few simple precautions are necessary:

1. Remove the ridge from the top of the cylinder before removing the piston to prevent breakage of the ring land on the piston caused by striking on this ridge.
2. Determine whether the ring grooves in the piston are worn and whether the wear is excessive. Check to determine whether the sides of the ring grooves are straight and free from uneven spots, nicks, or burrs.
3. Install only rings manufactured by reputable manufacturers of high-quality rings.
4. Fit the rings with proper side clearance for all the grooves and check in accordance with the clearance allowance specified by the manufacturer of the engine. Use proper but minimum end clearance for all rings on the piston.
5. When installing pistons in the cylinder, make sure that the rings do not strike the top of the cylinder block when the piston is forced in. Make sure that the rings are not damaged when they are closed in the ring clamp or other device used to get them into the cylinder.
6. Make sure that the engine operates without too much detonation and vibration. Detonation is closely related to ring breakage and piston fracture.

Summary of maintenance procedure. Consult the instruction book for exact procedure in all cases of maintenance. Take time to examine the rings and ring grooves, to measure the wear, and to determine, if possible, whether the troubles existing may be remedied. The following helpful suggestions may be considered:

1. *Rule for clearances.* Three kinds of piston ring clearances have been discussed: (1) the clearance between the ends of the rings at the gap, (2) the clearance between the ring and the sides of the groove, and (3) the radial clearance between the back of the ring and the bottom of the groove. A general rule has been proposed that the gap clearance for fire and compression rings should be from 0.005 to 0.0075 in. per inch of ring diameter. This indicates a clearance of 0.50 to 0.075 in. for a 10-in. diameter cylinder. This may be checked with the data given in previous chapters for slow-speed and high-speed engines. The gap clearance for step-cut and two-piece rings is usually somewhat larger. When rings have the overlapping ends, ring gap clearance is not so important. Rings for oil

control located below the piston pin can be fitted with gaps as small as 0.003 in. per inch of diameter.

The gap clearance can be determined by inserting the ring in the smallest or least worn part of the cylinder bore and measuring the gap by means of the feeler gauge. Remove the excess metal by filing off until the proper gap is obtained.

2. Effect of insufficient side clearances. When the side clearance of the ring is insufficient, the ring cannot adjust itself to the movement required to perform its function in sealing the bore, and will become carbonized and subject to sticking. Excessive clearance, on the other hand, especially in the 4-cycle engines permits the ring to hammer against the groove lands. This occurs on account of the reversal of forces between the exhaust and intake strokes; the inertia of the ring causes it to leave the land alternately during the period of the cycle. The rings lift from the bottom of the land groove and click against the top, an action that can sometimes be heard by careful listening near the ring zone. When the clearance is great, the movement will be heard as an intense hammering. The result is worn shoulders in the groove. Special designs and operating conditions cause variations from the rule for clearances. Generally speaking, greater clearances are permitted for 2-cycle engines, since 2-cycle rings usually do not leave the bottom land of the groove. For any particular engine, the instruction book gives the exact data for clearance.

3. Rule for radial clearance. The rule for compression rings generally requires that the radial clearance should not be less than 0.003 in. per inch of piston diameter, and not less than 0.012 in. total for pistons of 4-in. diameter and below. More radial clearance is usually permitted for oil-control rings—not less than 0.006 in. per inch diameter, but not less than 0.032 in. total for pistons of 5-in. diameter and smaller.

Checking diametrical tension. The force exerted to pull across the ring diameter, taken at 90 deg from the gap, to close the gap to its proper size or clearance is the diametrical tension. This can be determined by standing the ring upon a scale platform, with the gap at the horizontal diameter, then adding weights to the top of the ring until the gap closes to the fitted clearance. When the weight of the ring is subtracted from the scale reading, the result is the diametrical tension in pounds. Diametrical tension may be specified, but the "tangential tension" should also be known. This latter is the pull required on the ends of the ring to reduce the gap to the fitted clearance.

This pull can be determined by winding a wire around the ring, attaching a spring scale to the loose end, and pulling on the other until the gap clearance is correct. Correct ring tension will vary with ring thickness and diameter. For example, the diametrical tension for one ring of ½-in. face and 10-in. diameter was found to be between 55 and 65 lb, while another ring of ⅝-in. face and 10-in. diameter had a tension of 90 lb.

In the ring maintenance and inspection reports should be kept records of the original tension, as well as all clearances, the length of the diameters taken through the gap and at right angles to the gap, measured when the ring is pulled up to the fitted gap clearances. Keep a record of these various measurements to compare with tests when inspecting rings at the maintenance period when new measurements are taken.

Loosening stuck rings. Stuck rings may be reclaimed when tests indicate that the wear and ring tension are acceptable. When rings are so badly carbonized and stuck in the grooves that special effort is necessary to remove them, check for tension. The piston should be removed from the engine; it should be washed with a solution of one pound of lye to three gallons of boiling water. When the rings spring out free, wash off the solution immediately, and remove the rings. If the rings remain stuck, the piston may be soaked several hours in the solution. When any ring is so badly stuck that it cannot be removed or loosened, it is necessary to break it out. Sometimes the ring groove is damaged unless care is used in breaking it out. Screw drivers and chisels are dangerous tools for this purpose; use wood or brass. When the ring is loosened, the ends should be separated until a thin strip of brass or steel can be inserted under the ring to work it over the piston. The rings removed should be cleaned and any burrs removed with a smooth file. Check the rings on a face plate to determine whether they have been strained or warped into a spiral shape. The measurements for tension should then be compared with data for new rings.

Fitting rings to the grooves. Great care must be taken to make sure that ring grooves are true. When there are any shoulders in the grooves or any wear that makes the grooves out of true and parallel, the piston must be set up in a lathe and the grooves turned true. Each ring should be tried in its own groove by rolling it, face in, completely around the piston. The radial clearance must be checked at various points by placing a straight edge across the groove and inserting feeler

gauges between the ring and the straight edge. The side clearance is also checked by a feeler gauge. If an obstruction or binding is indicated, correct it before installing the ring. When the rings and grooves are cleaned, the use of a cloth, and not waste, is recommended so that no lint is left on the metal surfaces. Apply a thin coating of lubricating oil to the ring, work it over brass or steel strips to its proper groove. The ring should be installed with its proper side up and in the groove in which it was fitted to work. After rings are in place, again check to see that each is free in the grooves. Use proper method to enter the rings in the bore as the piston is lowered.

Well-fitted piston rings, free in their grooves, with correct clearances, are prerequisites to good engine operation, low maintenance cost, and good fuel economy. Poorly fitted rings usually result in increase in blow-by, impaired cylinder lubrication, and these in turn accelerate wear of the liner. The rings carbonize, liners are scored, compression is lost, contamination of the lubricating oil occurs, and loss of power is noticed. All of these troubles may be directly due to improperly fitted rings. When rings are properly fitted, then other causes for troubles with rings may be explored.

Summary and check list of ring sticking. When piston ring problems must be dealt with, a considerable number of related facts and causes that contribute to this problem will require careful study and a survey of engine operation in general. The following serves as a guide and check list for the procedure necessary to locate the causes of the trouble:

1. Blow-by
   a. Due to piston distortion, which in turn may be due to (1) overheated piston or (2) misalignment
   b. Cylinder distortion, which may be caused by insufficient cooling-water circulation
2. Rough or scored cylinder walls
   a. Worn cylinders
   b. Worn or distorted pistons
3. Worn piston rings
4. Excessive top piston land clearance
5. Weak piston rings, lack of tension
6. Insufficient ring groove clearance
7. Improper quality or grade of lubricating oil
8. Carbon from combustion troubles

9. Dust from the air accumulated in the combustion chamber
10. High piston ring temperature caused by:
    a. Engine operating at overload
    b. Incorrect lubricating oil viscosity
    c. Insufficient lubrication
    d. Insufficient jacket or piston cooling
    e. Lack of cooling water
    f. Scale in water jackets giving high jacket temperature
11. Defective valve action due to
    a. Warped valves
    b. Sticking valves, and so on.

Sometimes when an oil ring becomes filled with deposits so that the oil control is no longer adequate, the large quantities of lubricating oil reaching the compression rings cause them to stick in their grooves, and excessive blow-by sets in. The deposits in the oil rings occurred in the first place on account of blow-by caused by cylinder distortion if an improper liner installation was made. This permitted the hot gases to reach the control rings, solidifying the deposits constantly present at the oil control rings, burning the oil, and thereby making additional deposits, which rapidly filled the oil-control rings.

REFERENCES

Piston Ring Handbook, Double Seal Ring Co., Fort Worth, Texas, 1940.
Piston Ring Handbook, American Hammered Ring Co., New York, N.Y., 1945
Diesel Engine Operation and Maintenance, Texas Company, New York, N.Y., 1945

QUESTIONS

1. Name the fundamental functions of the piston rings.
2. Why must the piston ring have flexibility?
3. How is flexibility built into the piston ring?
4. What are the methods of imparting tension to the rings?
5. What causes the piston rings to exert frictional pressure?
6. Describe the labyrinth principle.
7. How does the gas pressure affect the wall pressure of the piston ring?

8. Why are a number of rings used, when one or two hold the compression?

9. What is the function of high polish and its relation to the proper seating of the ring?

10. What is side clearance?

11. How is side clearance determined or measured?

12. What causes the ring to hammer out the ring groove?

13. What causes the shouldering of the bottom of the groove?

14. What is to be done when there are shoulders worn in the ring land?

15. Why cannot rings function properly when the ring groove is shouldered?

16. What causes stuck rings?

17. What follows when rings do not seat properly? How is this detected?

18. What methods are recommended to test ring leakage?

19. What is the evidence that the rings are stuck?

20. When should the test for ring leakage be made?

21. How are hard-stuck rings removed from the piston?

22. Explain the method of testing the flatness of rings.

23. What is the procedure in installing piston rings as to inspection and smoothing piston ring edges?

24. Why should the rings be cleaned before installation, and what is used to clean them?

25. How can ring distortion during installation be prevented?

26. What is the bottom clearance and how much is necessary?

27. What is the effect of insufficient clearance in the bottom of the groove?

28. What is end butt or joint clearance?

29. How is suitable end clearance determined for plain rings, for bevel cut rings, and for step-cut rings?

30. What are some of the methods of reducing gap leakage?

31. What are some of the designs used to reduce gap leakage?

32. Does the installation of special rings require special attention as to which side is down?

CHAPTER 9

PISTON AND LINER MAINTENANCE

IN THE previous chapter, the related factors concerning the efficiency and the lubrication of the piston and rings were mentioned. A further study of these factors in connection with pistons and liners will help the operator to understand the application of methods now widely used to cope with the problems of maintenance.

Aluminum pistons are used in many of the high-speed Diesel engines and, to some extent, in the medium heavy-duty types. Experience with aluminum pistons has accumulated and the designs are highly developed. The advantages of aluminum pistons, however, can be offset by drawbacks in the design of the engine itself or of its combustion system.

The use of special materials in pistons and liners instead of cast iron has rendered the operation of the Diesel engine more efficient and reliable, and many modern designs have been completely worked out and developed. The operating engineer and maintenance expert is concerned with engine performance in the field, but it is a help when he understands the influence of these various features of design upon the operating condition and performance. It has been shown that engines are now built with cylinder liners made of metal other than cast iron, chiefly to reduce wear and to secure better cooling and more efficient lubrication. There is also an increasing use of chrome- and nickel-alloy steels for this purpose, and greatly increased efficiency and added endurance have resulted.

Heat stresses. The piston, cylinder, and liner are subject to severe heat stresses imposed by pressure and temperature. These stresses, resulting from power pressures and friction, are the most severe of any in the engine, and the working

conditions involved require corresponding study and understanding. These parts cannot receive perfect lubrication, on account of the high temperature and exposure to the products of combustion. *The operation is never ideal.* The adverse factors and influences are now enumerated.

Heat stagnation. Whenever the heat is not eliminated as rapidly by cooling as it is absorbed by the piston, the engine ceases to be an efficient and reliable machine. The total heat received by the piston must be dissipated to the cooling water

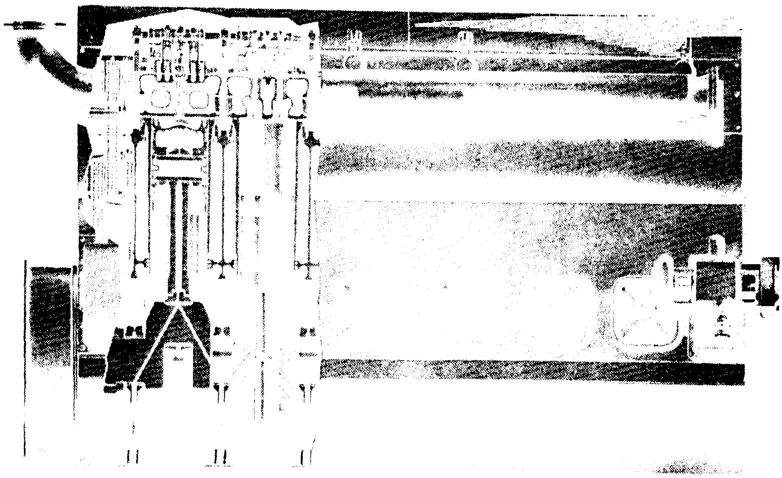


FIG. 9-1. Cross section of Diesel engine, showing wet-type liner, oil-cooled piston, marine-type connecting rod, force-feed lubrication, and overhead valves.

through the rings and by radiation. When there is any part of the piston where heat stagnates or accumulates, this part rises rapidly in temperature, excessively expanding the metal in the hot area, a condition usually followed by seizing of the piston. The formation of carbon deposits on the piston and the accumulation of scale in the water-cooling jacket will cause pistons to overheat. If for any other reason the piston becomes thermally overloaded, scoring of the liner, distortion of the piston, fractures in the crown, and complete or partial sticking of the rings result. Moreover, there is a lowered volumetric efficiency and falling off of the capacity of the engine to develop its rated load.

Distortion. On account of the abnormally high heat and the resulting expansion, the forces exerted by the piston, and the wedging action of the piston pin, all pistons are distorted; and high spots are found on the surface of the piston, which prevent the piston from bearing completely against the cylinder wall. The excessive pressure on these high spots eventually breaks down the film of lubricating oil, permitting metal-to-metal contact. When these high spots are small, they may eventually wear down to the surface level and polish out; however, when the heat cannot be dissipated sufficiently, the metal temperature rises quickly, burns or thins the lubricating oil, and destroys the film. Additional heat stagnation causes swelling and "bulging" of the piston. When only these high spots carry the friction load, with other parts of the cylinder not carrying their proper bearing load, the high friction in the piston contact is also converted to heat, and the temperature is soon high enough to cause seizure.

Seizure. It is evident that a very bad or severe seizure can occur in a very few revolutions because the mechanical energy available for conversion to heat is very great. Whenever the piston once starts to seize, it may freeze in place quickly and so solidly that great effort is required to remove it from the cylinder. A great deal of hard work with a powerful jack is then in order. The metal surfaces have become so highly heated that they are practically welded together by the heat at the moment of seizure. These seizures not only end in a "frozen" piston, but may also twist the crankshaft and bend the connecting rod.

A piston about to seize will be indicated by pounding and heavy laboring in the cylinder. The speed of the engine may commence to drop under the load, and this dragging will cause smoke. The engine should not ordinarily be shut down when about to seize, for this would result in freezing the piston. Instead, the load should be thrown off immediately when this condition is suspected. In the marine engine, the speed should be reduced to the lowest point that would keep the engine turning over, the fuel should be cut off the affected cylinder, the exhaust valve blocked open if possible, and the supply of lubricating oil increased on that cylinder as quickly as possible. To avoid freezing, the cooling water should be reduced for the affected chamber until it begins to boil to steam. This permits the piston and cylinder to expand evenly, allowing the piston more clearance to work more freely. After the engine has been

operated in this manner for a few moments, the affected cylinder will cool off so that the engine can be stopped, and the piston pulled for inspection.

Repair of stuck pistons. Should the piston not be distorted too much and the liner not too severely scored, they may be repaired. Considerable time to do the work is required as well as experience and skill. The roughened areas and spots can be smoothed down with a carborundum stone with rounded corners. The finishing is usually done by means of a smoothing file. This operation is too often useless; the proper remedy is to rebore and refit an oversized piston or replace the liner and piston. The part of the liner and piston so smoothed by filing will be below the surface, and if in the zone of ring travel, the rings do not function over this repaired spot and lubrication is more difficult.

It should be evident that when the high spots break down the film of lubricating oil and liberate the excessive heat, which cannot be dissipated to the cooling water from the affected area, the piston will probably *score the liner*. The cast iron swells and the structure becomes granular and scores away. The rings drag heavily on the cylinder wall, permitting engine output to be converted to heat by the friction at the high spots. The engine pounds, knocks, and the power falls off. The piston starts to seize because there are high spots opposite each other, one usually bulging first and forcing the piston out against the other side, so that it bears so hard that another high spot breaks through that side and scores it also. The piston is then wedged between the two spots on either side of the cylinder, a condition which can cause seizing or freezing, or even *the stopping dead of the engine*. This should be avoided at all costs.

New pistons. Replacement pistons should be secured from the engine builder. Suitable pistons for Diesel engines may sometimes be obtained from repair shops that make the castings themselves. It is not permissible to depend upon an outside manufacturer for these parts. In many cases, the pistons are shipped by the engine builder for refitting to bored cylinders. These are in the rough or semifinished form and are turned down to proper size for the rebored cylinder.

When fitting new pistons, especially to rebored cylinders, the working clearances should be determined accurately and carefully checked when the pistons are installed. Running in new pistons and rebored liners involves risks unless the

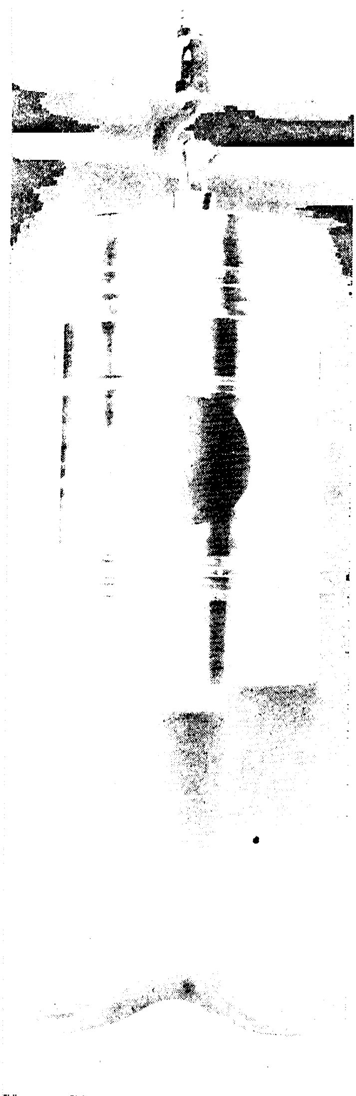


FIG. 9-2. Marine-type connecting rod assembly of forged steel, with small end bronze bushed. The box is of extra rigid cast-steel construction; four alloy-steel fitted bolts hold the box to the foot of the rod.

operator is experienced with the operation. If the liners and pistons have been properly fitted, and the surface of the cylinder is smooth, the engine should "run in" without undue risk and difficulty provided the operator observes caution when placing the load on the engine. A partial load should be first applied; a careful watch kept on the performance for some hours; and even after full load can be carried on the engine, it should have careful attention for some days.

Inspection of a piston while running in a new refitted job, or after the running-in, is very important. The piston should be pulled and examined carefully for distortion, local high spots, improperly fitted piston rings, and the color of the piston crown due to heat. A new piston, especially a green, improperly aged casting, will work satisfactorily for a few days during the running-in period. After that, the alternate cooling and heating begins to distort the piston, the annealing effect of the heat causes the piston to change its shape, or "swell," and sometimes cracks are found in the crown and in the ring belt area. If the casting was not seasoned or was not properly annealed after being cast, the piston will be green and therefore will warp. Such pistons eventually cause trouble by seizing, cracking, or scoring the liner. Pistons sup-

plied by the engine builder are seasoned and the castings annealed properly, but a repair shop casting may not be so carefully heat-treated.

However, if distortion is determined quickly after the running-in, the high spots can be machined down, the roughness and raised metal smoothed down with a file or emery stone to insure continued reliability in operation. This measure would be expected whenever it is necessary to cast the pistons locally. The rebored liners should also receive a careful inspection. When liners are rebored, they are held in a machine, which, on account of the strains of the heavy boring pressure, may distort the liner while it is being rebored. Therefore a liner may be out-of-round when released from the boring machine jigs.

Inspection of liner. Accurate measurements and a careful inspection of the liner before replacement will reveal any distortions due to reboring. This inspection should be made before the liner is placed in the cylinder or the cylinder is installed on the engine. The measurements are taken of the diameter of the cylinder in both directions—parallel and across the crankshaft. Commencing near the top where the greatest wear occurs, measurements with inside micrometers should be made at different intervals down the full length of the liner bore. This information should be charted.

Regular liner inspection and measurement should be made at the annual overhaul period, and if undue wear is taking place, the causes can be investigated. *More than 0.001 in. per thousand hours of operation for a large engine indicates a need for investigation.* Usually liners are replaced when the *wear has exceeded 0.005 in. per inch of cylinder diameter*, for example, 0.050 in. for a 10-in. bore. Modern practice has become more conservative, as indicated by previous instructions given.

Cylinder wear. The contributing factors of liner and piston wear may be more accurately determined when a careful record is kept. When the intimate history of the engine and the record of wear are kept, some progress may be made in associating the causes of wear. Reference has been made to piston ring records, inspection of the lubrication oil system, and so on. The wear on the piston and the cylinder should be known at all times by the operator as determined by inspection and measurements. Only in so doing can he develop proper judgment as to how to reduce wear and when replacements are required to improve operating efficiency.

It is considered good practice to pull pistons and inspect the piston and liner at definite intervals. This depends, to some extent, upon the conditions of operation and the age of the engine. Whenever the engine gives trouble and the rings are known to stick easily, the pistons should be removed, the engine inspected, and the carbon cleaned. Whenever this is

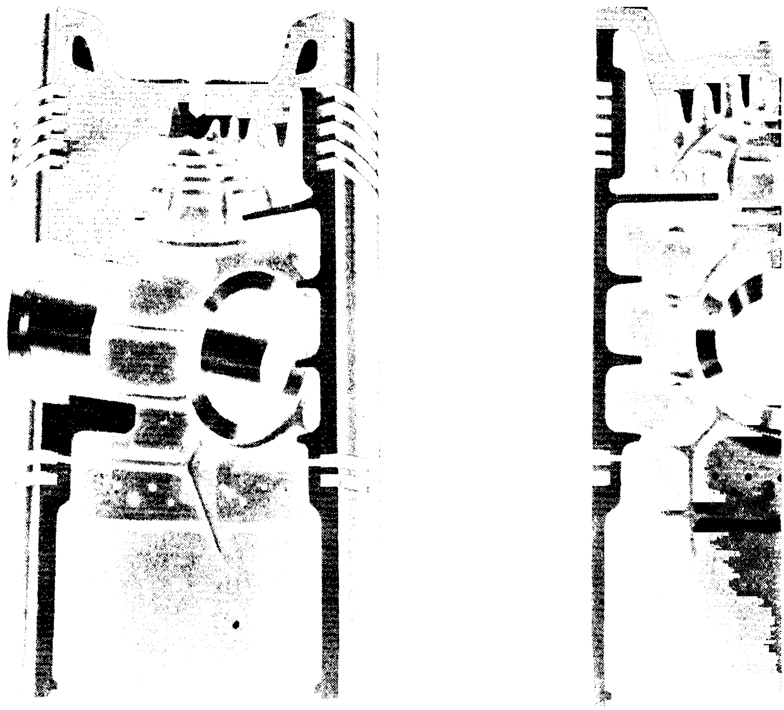


FIG. 9-3. Oil-cooled pistons. The piston heads are cooled by a jet of oil forced from the top of the connecting rods and impinging on the cooling ribs on the underside of the piston head. This helps to maintain correct piston temperatures and better lubrication of the rings and liners and reduces the possibilities of stuck rings.

required at short intervals, it is a definite indication that something is wrong with the operation or the engine or both. The practice of permitting engines to operate over a considerable time when it is known that rings have stuck in one or more grooves is not good practice. Rapid wear and low efficiency—that is, high fuel consumption—are more costly than repairs.

Measurement of the piston. Measurements should be made of the piston roundness and inspections should be made for distortion and for burned spots as well as for stuck and broken rings. Piston troubles are progressive; however, when timely inspections and corrections are made greater difficulties like sticking and seizure are avoided. Whenever pistons are pulled, the assembly should be cleaned of carbon, special effort being made to remove carefully the carbon from the ring grooves in the back of the rings. The counterbore in the upper end of the cylinder fills up with carbon and should be cleaned. Unless this is done, the lowering of the piston and rings back in the cylinder scrapes this carbon down ahead of the rings, wedging it at times between the piston and the liner wall, a frequent source of trouble such as scratching in the bore.

Every piston and ring assembly should be oiled as it is assembled. Avoid getting foreign matter on the parts. A moderate amount of oil should be poured around the rings as the piston is lowered and the rings should be likewise oiled in the grooves. Too much oil might prove dangerous if it burns when the engine is started, so care should be exercised and the surplus oil removed.

The top or crown of the piston should be inspected for cracks and minute fractures. When the cracks are small, they are usually not important even though they look dangerous. Unless the cracks extend into the metal, they may be disregarded. A test for the depth of the cracks is to moisten the top of the piston with kerosene, wipe it clean, and dust with powdered chalk. The kerosene in the cracks will discolor the chalk if there is an appreciable crack. This method helps to locate cracks in cast iron. A special magnetic powder will show more clearly cracks in iron and steel parts.

Measurement of cylinder liner wear. The average wear of the liner should be measured at four points parallel to the center line of the crankshaft and across the crankshaft. At the upper end of the travel of the top piston ring, the wear should be considered at a maximum when it is 0.004 in. per 1000 hr of operation.

Records of inspection. The deposits of carbon and lacquer on the piston will range from light to heavy. At the worst condition possible, all the area of the liner that is swept by the piston rings is definitely blackened, caused by blow-by of the gas, or coated with varnish, carbon or other decomposed

lubricating oil residues. The records should show the percentage of the area covered. This ranges from 0 to 100 per cent.

The pistons having all the skirt area below the rings coated with carbon, lacquer, and decomposed lubricating oil residues which are not easily wiped off with a cloth may be considered as 100 per cent covered. The percentage of the area covered and the location of deposit as well as its nature should be indicated as shown in the inspection form.

The pistons of 2-cycle engines need special attention on account of scavenging air-port restrictions. Deposits in both the inlet and exhaust ports should be inspected and cleaned when pistons are pulled.

Cylinder liner inspection also includes careful inspection of each engine part which may be found coated with sludge and carbon. A form for recording observations of deposits on engine parts is given below. The condition of the deposit is also indicated.

PISTON AND CYLINDER INSPECTION

Liner Deposits		Piston Deposits			
		Lacquer and Sludge		Lacquer Only	
Characteristic	Area %	Thrust	Antithrust	Thrust	Antithrust
Black					
Dark brown					
Medium Brown					
Light Brown					
Discoloraation					

Note: a. Piston deposits should be distinguished as sludge, lacquer, and sludge and lacquer.

b. The area covered by the liner deposits should be estimated accurately. Any spots which cannot be cleaned by wiping should be noted on a sketch.

c. Sample records of sludge and carbon should also be made a part of engine maintenance records.

DEPOSITS

Engine Part	Clean	Light	Medium	Heavy
Rocker Arm Box				
Rocker Arm Cover				
Push Rod Compartment				
Oil Screen				
Crankcase Oil Pan or Sump				

A sample of the sludge is taken and sent to a laboratory for analysis and report. The operator's records should also include an inspection record of the scavenging air-port restrictions and the frequency of cleaning as a part of each cylinder record.

DEPOSITS—SCAVENGING AIR-PORT RESTRICTION

Percentage of Restriction—Free to Completely Stopped Up

Cylinder No.	1	2	3	4	5	6	7	8	9	10
Inlet Ports										
Exhaust Ports										
Date Cleaned										

Advantages of cylinder wall finish. The three factors of cylinder wall finish have been discussed, namely, cylinder wall wear, distortion of the bore, and good finish on the surface of the liner. These mechanical characteristics are closely related. They are also related to cylinder wear and lubrication. Piston and ring wear is seriously affected when the hardness of the cylinder wall is increased unless the cylinder bores are finished to a very high degree of smoothness. A hard surface must be very smooth. This matter of hardness of liner material must be kept in mind when refinishing liners.

It was previously pointed out that the peening action of piston rings against the cylinder wall was responsible for much of the liner wear. This was indicated in the chapter on piston rings. The mirrorlike smoothness of cylinder bores

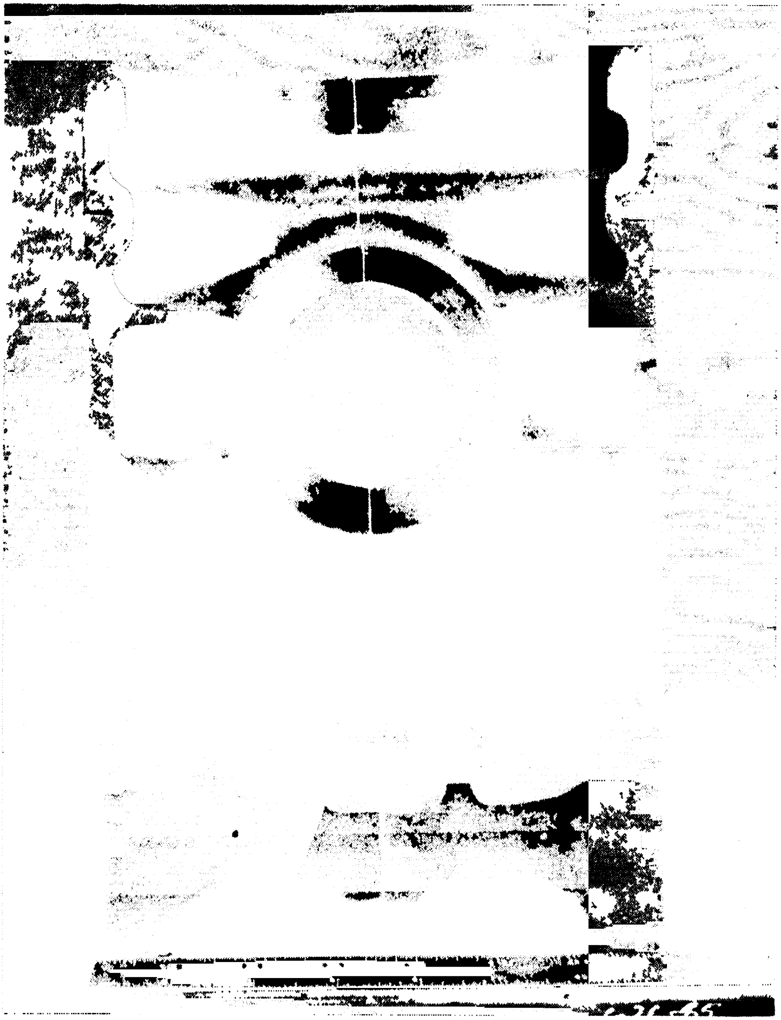
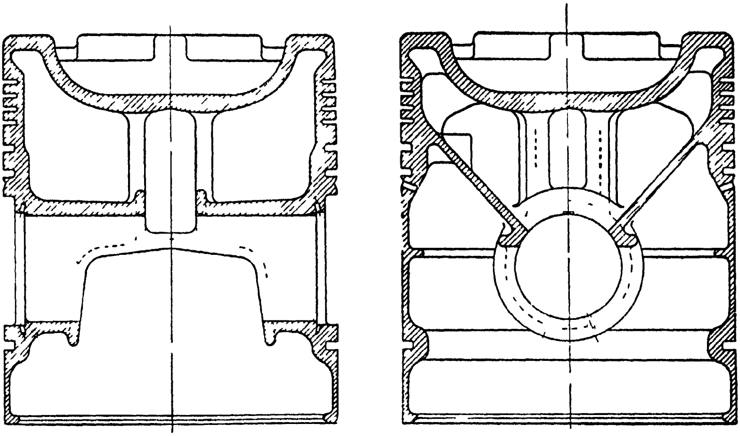


Fig 9-5 Cast-iron piston split to show construction This piston is of simple design, with four compression rings and two oil rings The head is designed for maximum heat dissipation

much as the power is increased, so the rate of heat rejection is materially less in proportion. In order to make a detailed analysis of this, Mr. Boyer reports a study of eleven engines, five of which were atmospheric and six, supercharged. This analysis is shown in Table 9-1. From this table it is seen that

the average increase in brake mean effective pressure or horsepower is 42.5 per cent. The over-all heat rejection, including lubricating oil cooler and water jacket, was 2482 Btu per horsepower-hour for atmospheric engines and 1841 for the



Courtesy of Cooper-Bessemer Corp.

FIG. 9-6. Example of efficiently cooled piston.

supercharged engines. The table therefore indicates that with a horsepower increase of 42.5 per cent the heat dissipation was increased only 5.85 per cent. This increase shows that the heat-dissipation equipment, with radiators or jacket water

TABLE 9-1

RATE OF HEAT REJECTION AND HORSEPOWER INCREASE OF ATMOSPHERIC AND SUPERCHARGED DIESEL ENGINES

	No. of Engines	Avg. Bmep	Increase in Hp (per cent)	Average Btu per Hp per Hour to Water	Bmep × Btu = K	Increase in Btu (per cent)
Atmos.	5	82.5		2482	205,000	—
Supchgd.	6	117 5	42.5	1841	217,000	5 85

coolers, need not be materially increased for normal ratings of superchanging. In fact, assuming that the heat-dissipation equipment is reasonably conservative in design, it should be adequate when changing over to supercharging. Water pumps present a similar situation since the rate of heat flow need not

be increased appreciably. The effect of supercharging on bearing pressures is included in Chapter 10.

One-piece cylinders. One-piece cylinders are made with the inner and outer walls of the cylinder cast in one piece. The advantage of this type of construction of cylinder is that the possibility of water leakage is eliminated; the construction, also, is suitable for small engines of the 2-cycle type because it eliminates the difficulty of sealing around the inlet and exhaust ports. One-piece cylinders also have sufficient metal thickness to permit reboring. The average heavy-duty engine, operating around 3000 hr a year, and using fairly good fuel and lubricating oil, will last 12 to 15 yr before the wear is sufficient to justify replacement. Under such conditions, reboring twice would give the cylinder a considerable life. Reboring a cylinder to an oversize means a new oversized piston. For small engines, a reboring job and an oversized piston cost very little more than a new liner. Some manufacturers accept the worn liner and piston in part payment for a rebored and oversize piston.

Precautions when installing a dry liner. When installing a dry liner, make sure that there will be a perfect contact between the inside of the cylinder bore and the outside of the liner. An imperfect contact interferes with the flow of heat through the two walls to the water jacket. This interference would cause overheating of the piston. Cylinders should be ground after being rebored and the outside of the liner should also be ground carefully to an oversize about 0.005 in. greater in diameter than the inside of the cylinder. If the liner is chilled by packing it with dry ice, it will contract sufficiently to go into place.

Separate liners. Many engines have separate liners with separate cylinders, especially if the engine has an en-block frame or cylinder box. The use of such liners applies to Diesel engines whether the engine is used in a stationary, portable, or marine application. Examination of the cross section of the engines with separate liners and an en-block frame shows that the cylinder liner of such an engine can be renewed without renewing the frame. With the cylinder-box construction, the liners can be renewed individually without replacing the cylinder box.

Advantages of separate liner. The separate liner has several advantages. The liner can be machined inside and out, a

procedure which insures accurate determination of the thickness. The cylinder, frame, or cylinder box may be of soft, close-grained cast iron, especially easy to machine, whereas the liner may be of harder material. Liners of chrome-nickel cast iron sometimes have a small amount of molybdenum added to the mixture. Hardness of over 220 Brinnel is obtained by accurate heat-treating. Hardness up to 450 Brinnel has been produced in liners of nickel or Ni-Resist cast iron. There is a general tendency to use somewhat softer liners of around 220 Brinnel for many applications.

Steel liners are found in England in small engines. There are some Americans who advocate steel liners, hardened by nitriding. The use of steel liners in this country has not made much progress so far.

One advantage of the separate liner is that it allows for expansion. The one-piece liner does not. The liner runs at a higher temperature than does the jacket. If the liner is of the one-piece type, the difference in temperature sets up strains or stresses in the casting. The wet liner is sealed at the top and bottom with the jacket to prevent leakage of the cooling water. The side and bottom surfaces of this fit are usually just machined to a snug fit. This is usually sufficient to prevent leakage. The top of the liner is grooved to correspond to a ridge on the lower face of the cylinder head. There is a copper gasket at the bottom of the groove, around $\frac{1}{32}$ in. thick. The use of this gasket in the groove protects it from exposure to the gases. The top of the liner is ground to the bottom of the cylinder head and a gasket is not used, the cylinder head resting on a shoulder. A jumper is used to carry the cooling water from the cylinder jacket to the head. The cooling water passes through the opening in the top of the jacket that corresponds with the opening in the bottom of the head. A metal tube instead of a rubber gasket is used here. Another type uses a rubber gasket around the outer rim of the jacket, with grommets around the bolts to prevent leakage of water around the cylinder head bolts.

Problems of rubber packing. Rubber rings and packing are stubborn when compressed, and great care is needed to make sure that the rubber has sufficient space. Liners may be pinched at the bottom because of lack of space for the rubber, and piston seizure may result. The rubber ring should be smaller than the groove. Rubber is not very good for sealing

around exhaust ports where the temperature is too high. A sliding fit at this point is preferred, or copper rings may be used instead of rubber.

Top cooling of liners. The severest cooling requirements are at the top of the liner, because of the high combustion temperatures at this location. Special provision for additional cooling at this point is now designed. On some engines, the shoulder supporting the liner is placed down from the top to eliminate concentration of metal around the combustion space. In this design, the water flows up in the jacket until it is level with the rim around the jacket from where it flows around the liner in a restricted path. Above the supporting shoulder is annular passage. The restricted size of the water passage causes the water to flow at a higher velocity, which is an advantage in cooling. Several engines employ this design to gain the additional velocity and to insure cooling at this point.

Two-cycle ports. On account of the high temperature at the exhaust ports, special provision is made for expansion and for prevention of piston seizure. This is done by thinning out the liner wall or "relieving" the liner at this point. Ports of most 2-cycle engines are well rounded at the corners in order to permit the piston rings to slide over the ports without catching or hanging.

Counterbored liners. The liners of some engines, particularly the larger engines, are counterbored. This counterbore is located so that the top piston ring moves part of the way into the counterbore. The purpose of this counterbore is to prevent the ring from wearing a shoulder on the liner. The depth of the counterbore is a little greater than the expected wear before the liner is replaced. Some small engines, however, do not have a counterbore, and hence when a shoulder is formed by wear, it must be removed, especially when rings are renewed.

Liner distortion. Rubber packing can distort liners, as previously noted. Excess flexibility in the frame of a small engine may cause distortion of the liner at the top, because the cylinder head studs, when tightened, pull the liner out of round. This permits the piston rings to cut and wear the liner bulge away. Pistons fitted with insufficient clearance will also cut into the liner.

Some causes of liner wear. Liner wear depends on a number of conditions, chief among which are these: kind of

fuel used, amount of dust in the intake air, kind of material in the piston rings, liner material, oxidation of the liner, frequency of starting, use of a poor grade of lubricating oil, and cooling efficiency. Fuel oil with a high ash content usually causes liner wear. High sulfur content is also a contributor to liner wear under some conditions of operation, especially when the engine operates for long periods at low loads. Piston rings that are too hard for the liner or forced too hard against the cylinder wall during the firing stroke will cause wear of the liner. Liners should be hard but not too hard. The oxidation, or burning, of the liner is most active when the temperature is excessive. This is the result of insufficient cooling. Liners also wear when scale is permitted to accumulate in the engine jackets. Starting the engine before the lubricating oil film has had time to form in cold weather makes for poor lubrication and promotes the wear of the liner; the more frequently this is done, the more the wear.

Liner wear is greatest at the top. Large liners especially become "belled" at this point, since this is the point of greatest heat concentration. Temperatures at the surface of the liner are as high as 1000° at the very top zone. The liner wear should be checked regularly and the rate of wear calculated. Measurements should be taken with a good inside micrometer both in the direction of the main shaft and in the direction of the crank throw. The taking of measurements at 4-in. intervals along the bore of the larger liner, and at smaller intervals for the smaller engines, is established practice. Careful records of these measurements should be made and preserved for reference. Care should be taken to make the measurements at the same place year after year. Normal or allowable wear is indicated in the wear-limit tables usually found in the instruction book. Normal liner wear is given by several authorities as 0.001 in. per 1000 hr of operation. Wear will be greater than this for the first 1000 hr and will become less afterward. Very few authorities agree on the amount of wear that calls for renewal of the liner, for reboring, or fitting dry liners. The engine builder, however, usually has some specific information in his instruction book.

Piston reclamation. When pistons are worn appreciably, it is sometimes possible to grind them down to true roundness and to install liners with correspondingly reduced diameters or bores.

Liner finishing methods. Boring the liner means taking a revolving cut with a hard-steel tool, which travels ahead along the internal bore as it cuts. It is now good practice to bore and then grind or hone the liner. The grinding is done with a rotary stone. An oil stone is used to hone the cylinder bore. A stone is made that both rotates and reciprocates. While grinding is the method of finishing large cylinders, the hone is used generally for the small bores. The smoother the interior of the bore, the better is the liner and piston lubrication. Lubricating oil fills the recesses and covers the high spots to minimize friction and to prevent scuffing and scoring. Grinding and honing produces a glass finishing superior to plain boring.

Scuffing and scoring. There is a distinction between the meanings of "scuffing" and "scoring." "Scuffing" is a minor scratch, while "scoring" is a serious cut, or series of cuts, deep in the metal. Any abrasions of the liner small enough to be designated as "scuffing" should become smoothed during operation. Serious abrasions may sometimes be removed by grinding the liner with a fine emery wheel and finishing with an oil stone. This frequently must be done when other remedies are not feasible. A serious score may be corrected by spraying on metal, although this must be done in a plant making a specialty of this service.

Deep scores in liners cause difficulties for the piston rings and prevent sealing off the combustion gases. Blow-by may result. The lubrication of liners with deep scores, even though ground out, is difficult because of the blow-by and the blowing of the oil off the surface of the liner.

Removal of liners. Liners should be removed in accordance with the specific directions in the instruction book. It is necessary to remove the head, piston, and rod. Lubricator connections must also be removed. The engine must be drained of circulating water, the lubricating oil drained off, and general preparations made for the removal of the liner itself. Procedures and details differ with every engine.

After the liner is removed, the water jackets must be cleaned of scale and deposits. The top and bottom fits of the cylinder jackets must be filed smooth of burrs. If the old liner is to be reinstalled, it must be cleaned of all scale and burrs. Ridges or shoulders in the top travel of the piston rings must be stoned out.

Fitting liners. Whether an old liner or a new one is to be used, it must be fitted to the supporting shoulders at the top

of the jacket by applying Prussian blue to the bottom of the shoulder on the liner, inserting the liner without any gasket or rubber rings, and turning it through a small angle either way from its position. The liner is then removed and the shoulder in the jacket examined for high spots, which will be shown by the Prussian blue. Such high spots must be scraped down, a small amount at a time, repeating the procedure several times until a good seat is obtained. Unless the seating is checked in this manner, the liner will be distorted when it is installed and the cylinder head tightened up. If the liner is sealed at the top with a copper gasket, a new gasket should be used.

The liner is then ready to be lowered into place. When the old liner is being reinstalled in the old position, the markings must be in line. When the liner is new, or an old one is being installed in a new position, it should be lowered into place as fitted by the method described above.

Causes of cylinder liner wear: check list. Definite causes for liner wear should be determined and recorded for the purpose of guiding the maintenance program. The following is a list of known factors which contribute to liner and piston wear. It serves as a check list when making a study of the inspection records.

1. Abrasive material in the fuel oil—sand, grit, and so on.
2. Abrasive substances in the lubricating oil.
3. Abrasive material in the air intake.
4. Water in the fuel oil.
5. Water in the lubricating oil.
6. The use of corrosive fuel.
7. Improper oil viscosity.
8. Overheated engine.
9. Insufficient, or lack of, lubricating oil.
10. Piston distortion and cylinder distortion.
11. Blow-by and piston ring troubles.
12. Frequent cold starts or jacket water too cold.
13. Excessive piston clearance.
14. Stuck piston rings.

Whenever wear is excessive, the cause or causes for the condition should be determined. This involves a study of the foregoing factors that contribute to the condition. These topics are referred to throughout this book. Refer to the index for numbers of pages where additional information concerning these topics may be found.

REFERENCES

"Heat Treated Gray Iron Cylinder Liners," by W. Paul Eddy, Jr., Metallurgist, General Motors Truck Co. Paper presented at the 1934 Convention of the American Foundrymen's Association, reprinted for the International Nickel Company, Inc., New York.

QUESTIONS

1. What piston and cylinder troubles are due to improper cooling?

2. What are the main causes of piston overheating?

3. Should replacement pistons always be obtained from the manufacturer of the engine? Why?

4. What are the precautions to be observed when fitting pistons and liners?

5. After the piston has been run in after refitting, what inspections should be made and at what intervals?

6. Why is the inspection after refitting pistons so necessary?

7. Why does a "green" or new piston casting usually cause trouble?

8. Why should the piston rings be inspected after running in for a period following installation?

9. When a rebored liner is being used, what precautions are necessary after installation?

10. What are the steps in inspection of the liner before replacement?

11. Describe the method of making measurements of and recording cylinder wear.

12. When and at what intervals should liner wear be checked?

13. What is a dangerous rate of liner wear?

14. When should liners be replaced, and what is the guide to determination of this need?

15. Give the main causes of liner wear.

16. What are some of the things that contribute to liner wear?

17. What is the proper practice with regard to pulling pistons and clearing rings?

18. When frequent cleaning of the rings and carbon from the piston is necessary, what is indicated?

19. Should an engine be operated when it is known that rings are stuck and the piston coated with deposits? Why?

20. Explain the procedure when inspecting the cylinder and pistons for deposits.

21. Name the kind of deposits, and the characteristics of such deposits, found on pistons, liners, and behind the rings.

22. When the pistons are reassembled in the cylinder, what are the general precautions and steps to avoid trouble?

23. What care should be taken in oiling the piston and rings when the piston is being placed in the cylinder?

24. What inspections should be made of the top of the piston, and how is this done?

25. How is the depth of the cracks determined?

26. What inspections of other parts are made when pistons are pulled?

27. What parts of the 2-cycle engine are cleaned of carbon when the pistons are pulled?

28. What kinds of deposits are found in the cylinder and on the piston?

29. What should be done about heavy deposits in the piston ring grooves? What does it indicate?

30. What causes the great heat stresses in the piston?

31. What and where is the distortion usually found in pistons?

32. What usually can be expected to follow piston distortion?

33. What causes piston seizure?

34. What are recognizable indications of piston seizure?

35. What are the steps to be taken when piston seizure is about to take place?

36. What repairs can be made on stuck pistons, when not too severely damaged?

37. What are the definite signs of cylinder distortion, and where would these indications be found?

38. What are the advantages of aluminum pistons?

39. What metal other than cast iron may be used for cylinder liners?

40. What are the precautions regarding piston to cylinder clearance, and where is this clearance information found?

CHAPTER 10

BEARING FAILURES AND MAINTENANCE PROBLEMS

THIS chapter discusses common Diesel engine bearing operation and failures and sets forth methods of progressive inspection of bearing failures. The object is to indicate reliable bearing maintenance methods. Routine procedure for general bearing maintenance was outlined in Chapter 6 as a part of the basic progressive maintenance. The use of a report on teardown inspection to include measurements of wear was also shown. This form indicated that every periodic bearing inspection should be recorded on forms prepared for the main and connecting rod upper and lower bearing shells. Such a record should contain related information such as date of inspection, bearing number, engine and bearing hours, lubricating oil used, and a description of any surface irregularities observed, such as pits, cracks, grooves, scratches, wiped and hard areas, embedded materials or foreign matter. When bearing failures are encountered, the record of inspection should be supplemented with sketches, photographs, or samples of the material, which might become very useful if subsequent bearing failures occur.

Inspection of bearing failures. All routine bearing inspections, especially those of defective bearings, must be made as thoroughly as possible in order to furnish data for the study of the failure and for making such adjustments and corrections as will decrease or eliminate such failures. Sketches of the bearing showing the locations of lost lining, cracks, pits, corrosion, and other surface troubles should be made, or photographs should be taken of the damaged bearings. Essential operating data should also be tabulated and studied in connection with the analysis of the bearing failure.

Diesel bearing failures. Experience has shown that the majority of Diesel engine bearing failures may be caused by one or more of the following conditions:

1. Fatigue of bearing metal under high cyclic loads. Excessive maximum or peak pressures may result in bearing failure.
2. Excessive or insufficient hardness of the bearing metal.
3. Inadequate bond between bearing metal and bearing shell.
4. Corrosion of the bearing metal by the lubricating oil.
5. Errors and incorrect methods of assembly.
6. Foreign particles embedded in the bearing metal.
7. Miscellaneous, such as defective manufacture, wiping, and so on.

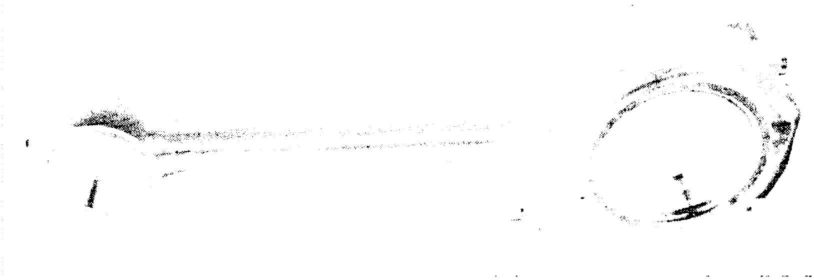


FIG. 10-1. Forged-steel connecting rod (Hendy Series 50 engine) made of high-grade steel and fitted with bronze wristpin bushing and precision-type shells in the crankpin boxes. An axial hole in each connecting rod carries oil under pressure to the full-floating wristpin and to the piston cooling jet.

1. Fatigue failures. Fatigue failures in heavy-duty engine bearings are usually caused by high cyclic loads as well as improper or loose fit of the shells in the housings. This means that a definite bearing area is pounded by the peak load every cycle, and if the physical properties of the bearing material, such as the compressive strength and brittleness, are inadequate, fatigue failure usually follows. Numerous small cracks, of mosaic pattern, on the bearing surface are the first indication of fatigue stress. With repeated poundings, fatigue cracks increase in number, finally penetrating the entire bearing material thickness, thus reaching the bearing shell. In such cases, sections of the bearing material may be lifted by the high hydrostatic oil pressure, resulting in a partial loss of the bearing surface. While not every bearing which shows fatigue failure and stress need be replaced, a bearing should be replaced if the

fatigue failure reduced its effective bearing area by more than 10 per cent before the inspection was made.

Bearings with fatigue failures, however, should be carefully inspected, and when it is considered that they may continue to operate, all sharp babbitt edges around the failures must be rounded by a scraper or a small electric iron in order to prevent any additional breaking off of the babbitt. Inspections of

FIG. 10-2. Precision-type removable main bearings (Hendy Series 50 engine). These main bearings are centrifugally cast, babbitt lined, and bronze backed. The journal diameter is 9 in. Each-bearing can be removed easily through the large inspection doors without disturbing either the crankshaft or the other bearings.

such bearings are necessary in order to ascertain that the failures are not progressing. After the bearing has thus been reconditioned and put back in service, an inspection should follow within a short time.

2. Hardness of bearing materials. The chemical specifications of several heavy-duty Diesel engine bearing materials are given in Table 10-1. Bearing materials should conform

closely to these specifications according to L. M. Tichvinsky of the American Bearing Corporation in a paper presented before the Baltimore Section of the SAE on February 10, 1944. The Brinnel hardness of the cadmium-base, copper-lead, lead-base, and tin-base bearing materials is plotted against temperature in Fig. 10-4. Since the Brinnel hardness is proportional to the compressive strength, this graph is doubly important,

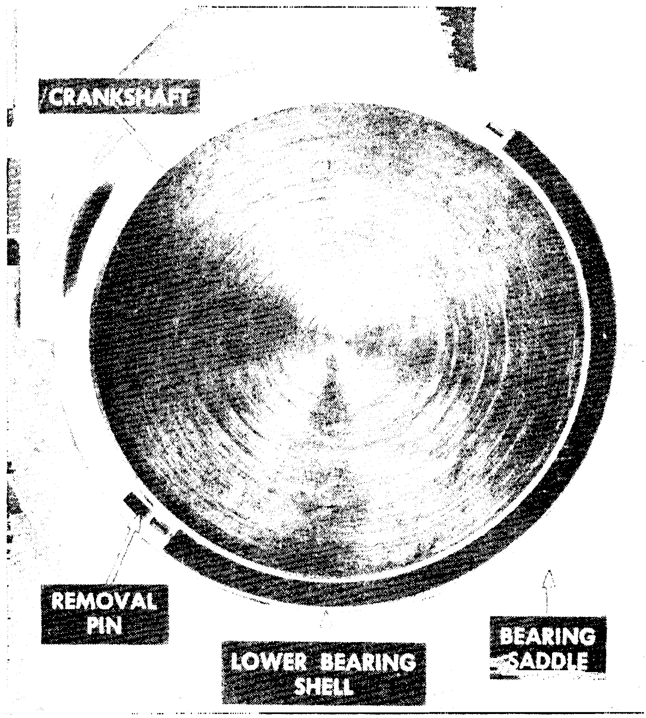


FIG. 10-3. Removing the main bearing shell.

Mr. Tichvinsky points out, indicating that the region of temperatures of particular interest is between 200 and 300° as shown by the double lines. The cadmium-base alloys are rather hard at room temperatures and rapidly lose their physical properties with increasing temperatures as shown by the steep curve. The copper-lead mixtures retain their hardness and physical properties almost unchanged in the region of temperatures plotted, as shown by the dotted line. In the group of white bearing metals, comprising Satco, tin-base and lead-base

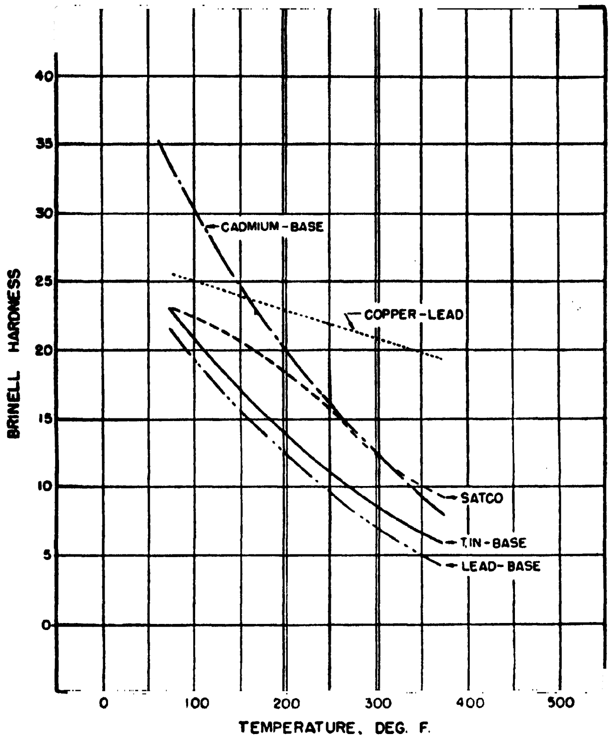


FIG. 10-4. Hardness-temperature relations of bearing alloys.

TABLE 10-1
CHEMICAL COMPOSITIONS OF BEARING MATERIALS

Bearing Material	Tin Sn	Lead Pb	Antimony Sb	Copper Cu	Cadmium Cd	Silver Ag	Nickel Ni	Arsenic As	Calcium Ca	Aluminum Al	Zinc Zn	Magnesium Mg	Iron Fe	Others
Cadmium-silver SAE No. 180	0.01	0.02	—	0.50	98.0	1.00	—	—	—	—	0.02	—	—	Rem.
Cadmium-nickel SAE No. 18	0.10	0.01	0.18	0.04	97.8	—	1.80	—	—	—	—	—	—	Rem.
Copper-lead	—	25.0	—	75.0	—	—	—	—	—	—	—	—	—	—
Lead-base Babbitt Navy Grade No. 6	5.00	80.0	14.0	0.50	—	—	—	0.20	—	—	—	—	—	Rem.
Satco	1.00	98.0	—	—	—	—	—	—	0.50	0.07	—	0.08	0.01	Rem.
Tin-base Babbitt Navy Grade No. 2	88.0	0.35	7.50	4.00	—	—	—	0.10	—	—	—	—	0.03	Rem.

babbitts, Satco possesses the highest physical properties at elevated temperatures.

This trimetal bearing is composed of a 0.006- to 0.008-in. thick layer of tin-base or lead-base babbitt bonded to approximately a 0.02-in. intermediate layer of bronze, which is in turn bonded to a steel back. This makes a high-grade bearing material.

When bearing material is too hard, it may score the journal surface, whereas material that is too soft usually yields under bearing pressures.

3. Corrosion of bearing alloys. The corrosion of bearing metals and materials is ascribed to the chemical action of oxidized lubricating oils and can be identified by the rough and pitted surfaces frequently observed on Diesel bearings. This effect of oxidation on bearings will be more fully discussed in the chapter on lubricating problems. In the majority of cases, corrosion over small areas of the bearing is associated with high localized temperatures. Corrosion may also occur over the bearing area as a result of a small amount of sulfuric acid that forms in the crankcase when the *sulfur trioxide* from the fuel mixes with water condensate while the engine is shut down.

Tin-base babbitt, which has a high corrosive resistance, is usually called a *noncorrosive babbitt.* However, under unusual and adverse conditions, it too may corrode. Corrosion of any bearing load-carrying surface is to be anticipated in inspections, for corrosion decreases the load-carrying capacity rapidly, once the chemical action sets in. When found, it may require replacement of the bearings, and whenever the effective bearing surfaces have been reduced to such an extent that wiping has occurred, replacement is indicated.

4. Bond strength between bearing lining and shell. Poor bond between the bearing lining and the shell metal is a rather frequent cause of bearing failure and, in most cases, is due to defective manufacture. The heat flow through a poor bond area is usually impeded and may result in failure because of uneven and excessive bearing surface temperatures. Fatigue cracks over poor bond areas will loosen the linings quickly. Inclusions or foreign matter in the bond are usually due to imperfect cleaning of the shell surface before applying the bonding alloy. In connection with the *trimetal type* of bearing mentioned above, poor bond may occur between the inter-

mediate bronze layer and the steel of the shell. Nonmetallic inclusions between the bronze and the steel will decrease the heat flow from the bronze to the steel, and overheating may result. Such manufacturing defects are not usually determined or detected by routine inspection and are usually found only after bearing failure. The areas of poor bond of a failed bearing are usually without any trace of the bonding alloys.

5. Improper or faulty assembly. Assembly errors do occur from time to time and are usually caused by negligence and lack of experience. Faulty assembly of a small bearing shell may occur because the locking lip does not fit the recess in the bearing housing, and thus it may produce distortion that leads to bearing failure. Another example of faulty assembly is that of connecting rod bearing mounting of the upper grooveless shell in the place of the lower shell, which is machined with an oil groove. This prevents the flow of the oil, and failure occurs almost immediately after the engine is started.

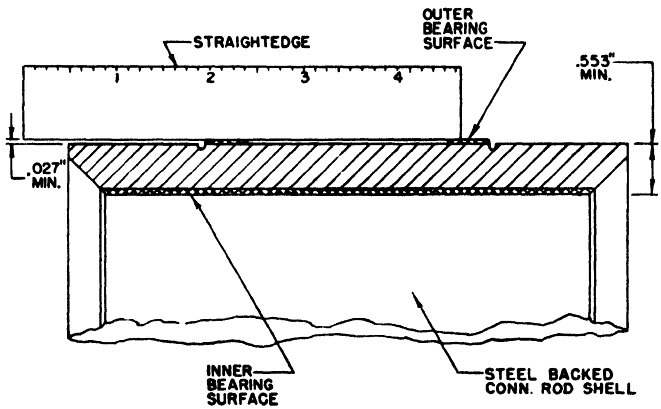


FIG. 10-5. Connecting rod bearing shell wear limits.

Another example of faulty assembly is the seizing of the outside surface of the upper main bearing shell to its cap. This kind of seizure, or local welding of small areas and transfers of metal, is caused by the working of the shell in its cap at the bearing split. This working is possible because of excessive side clearance, which may be found between the cap and the saddle when improperly assembled. Any excessive amount of lacquer on the back of a bearing indicates poor contact between bearing and cap. Poor contact may be due to excessive clear-

ances, foreign matter, and rough finish.* This also considerably decreases the normal heat flow, and failure may follow.

6. *Failures due to foreign particles.* Foreign matter, especially hard particles, may find their way into the lubricating oil line and into the bearing and thus damage by scoring both the journal and the bearing surfaces. Such particles will be embedded by soft bearing linings and rendered harmless. These white-metal bearing linings, 0.006 to 0.010 in., have higher load-carrying capacity than thick linings of 0.015 to 0.0125 in., and are now more frequently used than the thick linings. However, these thin linings cannot fully embed large foreign particles, and their presence will result in scoring the journal.

7. *Miscellaneous failures.* Careful inspection of a number of bearing failures shows that *intrusions,* localized areas of hard and porous babbitt, will cause initial bearing failure. In addition, there may be copper segregation with the possibility of scoring the shaft journal.

a. Wiping of bearing lining usually results from slight metal-to-metal contact of elevated lining or journal areas. The surface of the wiped lining is very smooth, approximately 3 to 7 microinches, while the surface of a well-machined lining varies from 20 to 50 microinches.

b. Local wiping on the top of the bearing is usually indicated by a slight elevated polished area produced by the displacement of superficial bearing lining. In such cases, the bearing should be replaced.

c. Wiping not necessitating bearing replacements is very common, especially during the initial period of operation. These bearing areas so wiped are not elevated over the bearing surface and are usually well-polished areas of indefinite, thoroughly oriented shapes. This condition can be identified by experience when bearings are being inspected during the original or initial running in of the engine.

Maximum pressures of Diesel bearings. The maximum Diesel engine bearing pressures deserve careful study and thorough understanding. The pressures are due to high cyclic loads and peak pressures. The operation of the piston pin, connecting rod, and main bearings of a Diesel engine differ

to a considerable extent, depending upon the type of engine and magnitude of loading.

1. Piston pin bearings. Maximum pressures are very high: they may run to 6000 psi. This high pressure is due to large peak forces imposed on small bearing diameters. The loading is due primarily to combustion and inertia forces. On account of the swinging or rocking motion of the connecting rod, the load acts over a small arc of the bearing circumference, approximately 15 deg. The lubricating oil temperature is rather high because it is usually supplied through the main and connecting rod bearings, successively. The piston pin surfaces are very hard, approximately 60 Rockwell C hardness at 150 kg load. These surfaces, often chromium-plated, are very smooth, below 10 microinches. The piston pin bearings are also designed with small clearances, and successful operation depends greatly on the dampening properties of the lubricating oil.

2. Connecting rod bearings. Maximum pressures in this case are also very high and may approach a value of 3000 psi.

a. The loading of the connecting rod bearings is due to gas pressure, inertia, and centrifugal forces.
b. The upper shell supports most of the load in a 2-cycle Diesel engine, while in a 4-cycle Diesel engine both the upper and lower bearing shells carry a considerable load.
c. Connecting rod bearings are better cooled than are the main bearings as a result of the ventilation produced by the rotation of the crankpin journals.
d. The lubricating oil is usually supplied from the main bearing, being forced through the oil groove and the inclined bores of the crankshaft.
e. Crankpin surface hardness varies from 200 to around 300 Brinnel and depends on the bearing material hardness.
f. The surface finish of the crankpins should be as smooth as possible or practicable.
g. These bearings are designed with clearance ratios ranging from 0.007 to 0.0015 in. per inch diameter. Smaller values are used in some 4-cycle Diesel engine bearings.

3. Main bearings. In the main bearings, the maximum pressures are moderate, with values usually under 1500 psi.

The loading of these bearings is due to gas pressures, inertia, and centrifugal forces.

a. The bearings are lubricated with cool lubricating oil direct from the head exchanger or engine sump.
b. Main journal hardness is the same as that of crankpins.
c. These bearings are designed with clearance ratios varying from 0.008 to 0.0015 in. per inch diameter.

4. Other Diesel bearings. Camshaft, gear, pump, and auxiliary machinery bearings, which operate as power bearings, are not heavily loaded and cause very little trouble when properly lubricated and when the engine is kept clean.

Diesel bearing maintenance. It should be obvious from the discussion of the bearing failures that there is no material that will satisfy and comply with all the requirements of a good bearing material. The selection of a bearing material, for this reason, is made by the designer, and it should result in the optimum combination of lining material, journal hardness, and lubricating efficiency.

The maintenance and performance of the precision-type bearings used in small engines, under 3-in. journal diameter, and in large engines, over 3-in. journal diameter, differ considerably. The manufacturing, including grinding of small crankshafts, is done with a high degree of accuracy, which results in uniformity of diameter and in small tolerances. Bearings of such small journals, which for uniformity are often diamond-bored, are interchangeable and do not require *hand-fitting*. The manufacture of large crankshafts, on the other hand, is not so accurate, resulting in large final tolerances. In addition, the surface finish of the large journals does not fit as smoothly as small journals. Bearings for large journals might not fit the desired area (70 per cent contact or more), and usually are hand-fitted by experienced mechanics.

Lubrication. The use of approved and *low-corrosive lubricating oils* is an important factor related to bearing maintenance. Adequate cooling of the lubricating oil will diminish the possibility of rapid oil oxidation. The usual practice now is to have a *chemical analysis* of the lubricating oil made at appropriate intervals, from 100 to 500 hr, depending on the service condition, in order to check for dilution and oxidation. Inspection of the sump oil samples, drawn from the lowest point, should be made for metal particles at regular intervals, together with

the inspection of the sump oil filter screen. Cleanliness in handling bearings, especially during assembly, is extremely important.

Bearing and shaft problems. Several factors, such as vibration of the shaft, the type of drive used to transmit the power of the engine, and high operating pressures and speeds, have a relation to bearing maintenance.

1. Wear of upper shell. While the weight of the piston and connecting rod is always downward, the shaft will wear the upper main bearing lining. This is due to the fact that there is little pressure opposing the motion of the piston on the exhaust stroke. The piston and rod start from a position of rest at the bottom and attain a maximum velocity at the mid-point of the stroke. From this point to top dead center position, the piston is retarded by the crankshaft against its momentum or tendency to maintain the maximum velocity against the retardation. This retardation or reversal of forces results in a force being exerted on the crankpin that slightly lifts the shaft and presses it against the upper bearing shell lining.

This problem of reversal of forces or vibrations applies to large heavy-duty engines having heavy reciprocating parts. It is evident that the total mass of these weights acting downward may not be greater than the force exerted upward by the retardation of the piston. If it is greater, the shaft would remain on the lower bearing shells; if not, the shaft lifts.

2. Vibrations. When an engine is connected to a pump or other power-driven machinery by means of a long shaft, the unbalanced pump impellers will sometimes set up a vibration in the engine shaft. There may also be *torsional vibrations* or the vibration of the shaft resulting from running in a slight twist. Engines should not be operated with such vibrations. Especially when rigid couplings connect the shaft to the engine, the shaft may vibrate. Flexible couplings are preferred if the driven machine or device imposes alternate or variable loads on the driver.

Repairing cracked babbitt. Unless the cracks in the babbitt are small and insignificant, repairing cracked shells is not good practice. When the cracks do not run the entire length of the width of the shell, the bearing shell may occasionally be repaired by grooving it with a hammer and a *cape chisel* or other suitable tool and running in fresh babbitt to fill the cracks and other small places where the babbitt may

be missing. When the whole shell is divided into sections, a rewelding job can be successful only when done by a skilled mechanic with experience.

Effect of worn bearing. Operating the engine with excessive bearing clearance has been discussed elsewhere. It must be added that the results of operating in this condition may be serious. Too much clearance permits the lubricating oil to leak at the bearing ends. This leaking is aggravated when the oil runs too hot and thins down to a point where the normal

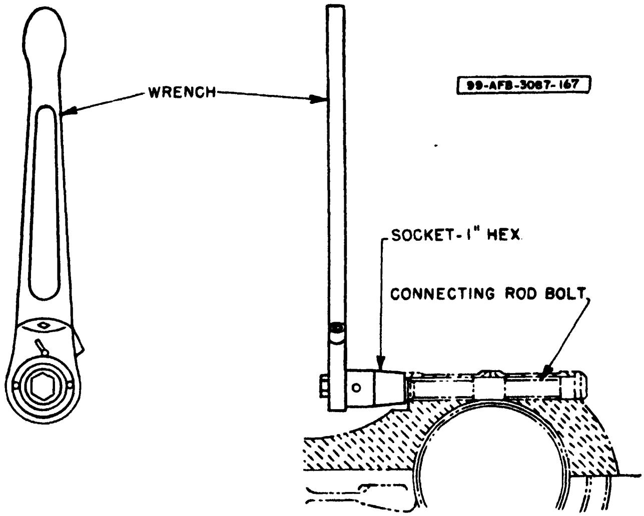


FIG. 10-6. Connecting rod bolt nut wrench.

clearance may permit the oil to escape. This matter of clearance is very important in the high-speed Diesel engine. If the clearance is large, the shaft will lift and fall as explained and permit hammering and pounding out of the bearing linings.

Out-of-line crankshaft bearings. Running bearings out of line causes rapid wear on these parts. The unbalanced forces also contribute to wear of the bearings. Unequal compression in the cylinders, or irregular or unbalanced firing pressure, is another possible source of much abnormal bearing wear. Too much advance in timing, which permits peak pressures before and at top center, should not be permitted. If these firing pressures are applied too soon, the added force on the crankshaft and bearings breaks down the oil film on the bearings by excessive bearing pressures. When metal-to-metal contact

occurs, wear is rapid. Any engine that pounds and knocks is likely to have bearing troubles follow. The working pressures in the cylinder and the engine balance should be carefully checked and adjusted. The indicator discussed elsewhere aids in keeping these forces down to a minimum.

The misalignment of the bedplate and frame, or a twisted or bent shaft, has been found to be the cause of bearing troubles. In the marine field, it has been noticed that the engine may have a tendency to spring out of alignment. Connection to propeller shafts also may cause vibration that affects the bearing wear.

Scraping bearings. It should be understood that scraping a bearing is not a job for an inexperienced mechanic. The high spots must be scraped until a good, smooth bearing surface is obtained. At least three fourths of the bearing surface should take up the Prussian blue when a bearing is fitted closely enough for use.

Requirements of the bearing metal or material. It has been shown that the durability of the bearing metal is important

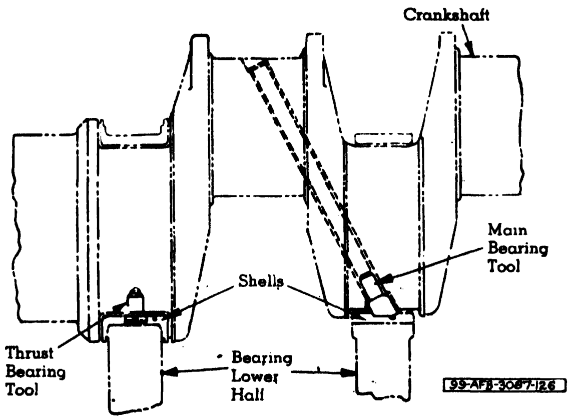


Fig. 10-7. Main-bearing shell removing tools.

as it must be tough and not so brittle that it will break up in service. The metal should be properly selected so that the kind used will have sufficient resistance to corrosion. The material should be the correct kind and composition for the type of engine and service. It must have sufficient strength to sustain the load imposed upon it. Only the very best resistant Diesel engine babbitt alloy should be used for wrist pin bearings. These bearings are usually centrifugally cast and the tin content

is high. The metal also contains some nickel in most cases. Some metal manufacturers use a secret process for which great advantages are claimed. The engine builders usually use well-known alloys of standard composition and employ bearing designs in which the operating and service risk has a history. Any departure from the manufacturer's recommendations in the instruction book is risky.

It should be understood that the causes for cracked bearings may not always be as easy to understand as may have been implied by the foregoing discussion. The extent of cracking of the lining that may be safely permitted before replacement is not always easy to determine. The condition of cracked babbitt may not be serious as long as the shaft is in proper alignment. Unless the cracks are very numerous, with pieces of the lining working loose, the shell may be retained and operated in this condition for some time. It is frequently found, upon teardown of an engine, that the bearing shells have been cracked for some indeterminable time. It has been possible to tighten up the cracks by pounding the edges together with a *ball peen hammer*. A risk of scoring the shaft is involved, however.

Effect of supercharging. When the supercharging process is added to the inlet air, an increase in the peak pressure results. The cylinder pressures of supercharged engines increase 10 to 15 per cent over the nonsupercharged of otherwise identical types. This increased pressure naturally reflects itself in increased stresses of all parts, including pistons, connecting rods, and bearings. These increased peak pressures are taken into consideration when designing engines that are to be supercharged, unless the engine was designed with this margin of safety before supercharging was added.

In many cases it has been found necessary to redesign the bearings for supercharging. Bearing pressures permitted on any given design of bearing is dependent in large measure on how long that pressure is sustained. Detailed study of the area of the pressure-time diagram of the bearing shows that frequently this area is somewhat less with supercharged engines than with the atmospheric type. This is because of the combination of more sustained load and inertia forces. An interesting study of this problem was discussed by R. L. Boyer at a meeting of the Cleveland Section of the American Society of Mechanical Engineers, Cleveland, Ohio, May, 1945. The polar

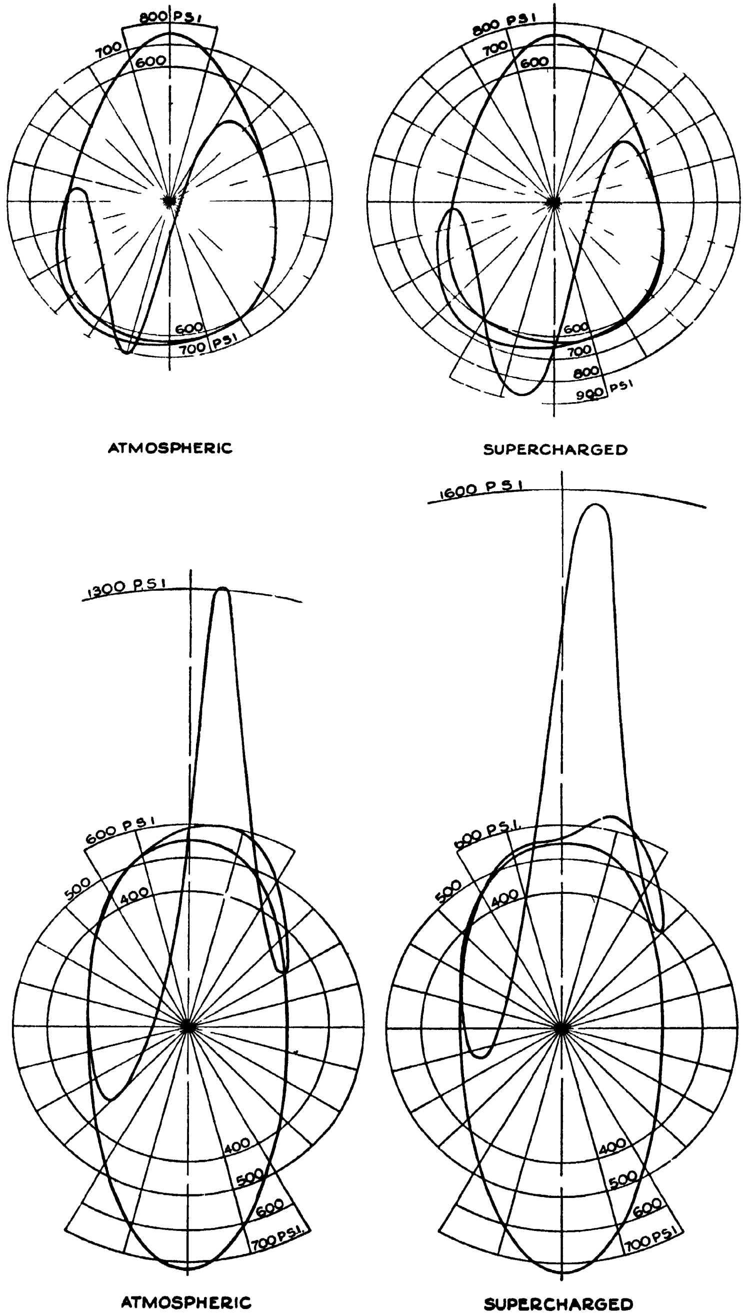


FIG. 10-8 Polar diagrams

diagrams presented with his paper are shown in Fig. 10-8. The diagrams indicate that, at zero speed, the pressure-time diagram has an area about in proportion to the appearance of these polar diagrams. Mr. Boyer points out that most modern Diesels, if they have precision-type bearings that have been properly designed, have adequate bearings to permit supercharging. He reported analysis of several models of engines showing relative duty of the atmospheric versus supercharged type.

Cleaning oil ducts and passages. Failure to clean properly the oil ducts results in many bearing failures that occur almost immediately after an overhaul. Whenever connecting rods are out of the engine, the oil ducts should be cleaned immediately, utilizing air pressure for blowing out any deposits and foreign matter. It is sometimes necessary to use a wire probe even when high compressed air or steam pressure is available for blowing out oil passages. It is a mistake to use waste instead of cloth when cleaning parts, since some of the waste may be left in the ducts and eventually lead to plugging of the passage.

The rifle-drilled oil ducts in the rods of some engines must be cleaned very carefully since these are depended upon to feed the piston pin bearings. The surface between the connecting rod and the bushing should be cleaned, and a snug fit between the rod and the bushing must be obtained. A perfect fit is necessary in high-speed engines, and good automotive practice should be followed. When bushings are properly fitted and seated in the caps and in the connecting rods, which may sometimes involve scraping the high spots on the back of the bushings, the edges must meet and form a smooth surface level with the cap and rod forging. Sometimes a file is used for smoothing the edges. An oiltight joint between the bearing halves of the rod when they are pulled up against the shims is essential, and when they are not pulled up, the oil pressure will be below normal or lost completely, owing to leakage at the joint of the bearing.

When the connecting rod and piston are replaced in the engine, if the rod is marked, the side marked "Front" should be properly placed. Check the clearance between the edge of rod and bearing and the crank throw. If it is too wide, a file is used, which requires careful work, or a refacing tool is used to face off the side of the bushing until there is a clearance of approximately 0.010 in. to leave a snug fit between the edge of

the bearing and the crank check. The sidewise clearance for a high-speed engine should never exceed 0.015 in. A blade gauge is used to measure this clearance. The rod is crowded against the opposite crank check and with the use of a pry bar, the clearance is determined by careful checking.

Types of crankpin bearings. While numerous types of crankpin bearings have been designed, present types may be divided into two classes, the "marine type" and the "cap type." In the marine-type rod, the top half of the bearing is separated from the rod itself. The rod ends in a foot to which the top and bottom half bearings are assembled. The assembly is bolted together by crankpin bolts, either two or four bolts being used. The foot usually contains a recess into which the raised part of the top box is fitted or which is *spigoted* into the foot. This prevents the box from slipping with respect to the foot and saves and relieves the bolts of the strain of withstanding a shearing action. The top half is also spigoted into the bottom half for the same reason. When the two halves are not spigoted together, shifting is prevented by the use of dowels; the halves of the bearing are separated by the shims to permit adjustment of the bearings. Shims are used between the top half and the foot also and permit shortening or lengthening of the rod. The crankpin bolts are not "fitted," but are thickened at the middle section where they pass through the adjoining boxes and also at the ends. The thickened parts are machined to close tolerance in order to fit very close at these points. Small cap screws are used at the bottom to keep the bolts from turning. The nuts also have locked recesses so that cotter pins can be used to lock them in position. Dowel pins are used in addition to spigoting to hold the bearing in position.

Oil grooving in the bearings. The inside of crankpin bearings have oil grooves through the center, around the circumference with the groove connecting with a hole in the connecting rod (Fig. 10-2). The lubricating oil is fed into the groove from a drilled hole in the crankpin to lubricate the crankpin bearing, then passes up through the rod to lubricate the piston pin bearing. Rods of the *H* section lead the oil up to the piston pin by the use of a tube fastened to the side of the rod. A ball check valve is used in the oil lead near the foot of the rod to prevent the oil from draining when the engine is not running. The check valve also serves to prevent the inertia force from driving the oil downward when the rod moves in that

direction. Some older engines used oil grooves lengthwise of the pin as well as circumferentially. These grooves did not extend to the edge of the bearings, yet much oil escaped through them, and for this reason, such grooving is no longer employed.

Methods of babbitting bearings. The babbitt is applied directly to the boxes in larger bearings. There are dovetail grooves in the boxes of some bearings so that the babbitt is anchored in place. The babbitt is cast directly on the boxes made of cast steel or of cast iron. The babbitt is relieved slightly at the joint between the halves, but not at the ends of the bearing, for this relief at the ends would permit the lubricating oil to escape.

Bearing shells. The babbitt may be cast in shells, which may be made of bronze or steel, and which may be of considerable thickness in some bearings and very thin in others. Shells are separated by shims if the babbitt is thick; but if the babbitt is very thin, shims are not used.

Cap-type rod and bearing. The foot of the cap-type of rod and the upper half of the bearing box are forged in one piece, with the bottom half serving as the "cap." This type does not make provision for removing shims between the foot of the rod and the upper-half box, but otherwise, there is little difference in the construction of the bearing. The cap-type rod is usually made somewhat lighter than the marine-type rod and, for that reason, is now used on most of the small engines.

General bearing metals. Babbitt is the oldest bearing metal in use at the present time. It consists of an alloy of tin, copper, and antimony. It may be soft, medium hard, or hard, depending on the relative proportions of the three elements used. A mixture of 88.9 per cent tin, 3.7 per cent copper, and 7.4 per cent antimony results in a soft babbitt; while 80 per cent tin, 10 per cent copper, and 10 per cent antimony results in a hard babbitt. Babbitt is also called "white metal."

Special bronzes are generally used for small engines. A bronze is made up of 73 per cent copper, 12.5 per cent tin, 10 per cent lead, and 4.3 per cent graphite, and will stand heavy loading. It requires very little lubrication and does not seize or score a steel pin. Copper alloyed with cadmium and silver has been used in the latest high-speed engines, as previously indicated.

Precision bearings. The use of thin bearing shells is a late development made possible by the development of the new bear-

ing metal alloys. The shell and the bearing metal are very thin and the precision finish is very accurate. The shell is made in halves and is kept in the bearing box by lugs to prevent rotation. The dimensions and sizes are so controlled that it is not necessary to fit such bearings. When the wear is sufficient, the bearings are thrown away and new ones replace them. Pins running in such bearings are harder than for babbitt but require a copious supply of lubricating oil of high viscosity and at good pressure running through them at all times.

When operating an engine with the modern type of bearings, it is absolutely necessary to select a lubricating oil that will not attack the bearing metal. A number of lubricating oils for high-speed engines contain "additives" for the prevention of ring sticking, and these additives frequently cause corrosion when used with cadmium-silver and high-lead bronzes.

Connecting rod bearing clearances. There is no general rule for proper bearing clearance under all conditions and for all engines. The recommendations of the engine builder as found in his instruction book should be followed carefully. In the absence of such information, a rough rule may be followed.

Maximum allowable clearances. The maximum clearance is usually given by the instruction book. Excessive clearance permits the engine to pound when starting and stopping, or when operating below rated speed. When 4-cycle piston pin bearings have too much clearance, the engine may pound during the middle of the intake and exhaust stroke. Clearances are usually adjusted before they exceed twice to two and one-half times the initial working clearances. Adjustable bearings are adjusted, but nonadjustable bearings are replaced when the wear exceeds the allowable limit.

Clearances in adjustable bearings are usually determined by means of the lead wire. The procedure was previously explained (Chapter 6).

Rebabbitting bearings. When bearings are rebabbitted, the first step is to first remove all traces of the old babbitt, scraping off as much as possible, then heating the shell to melt the babbitt out of the dovetail grooves. The shell is then thoroughly cleaned, first with gasoline, and then with caustic soda.

There are two recognized methods for rebabbitting bearings. One is casting the babbitt in the box with a mandril located in the position of the pin; the second, centrifugal casting, or

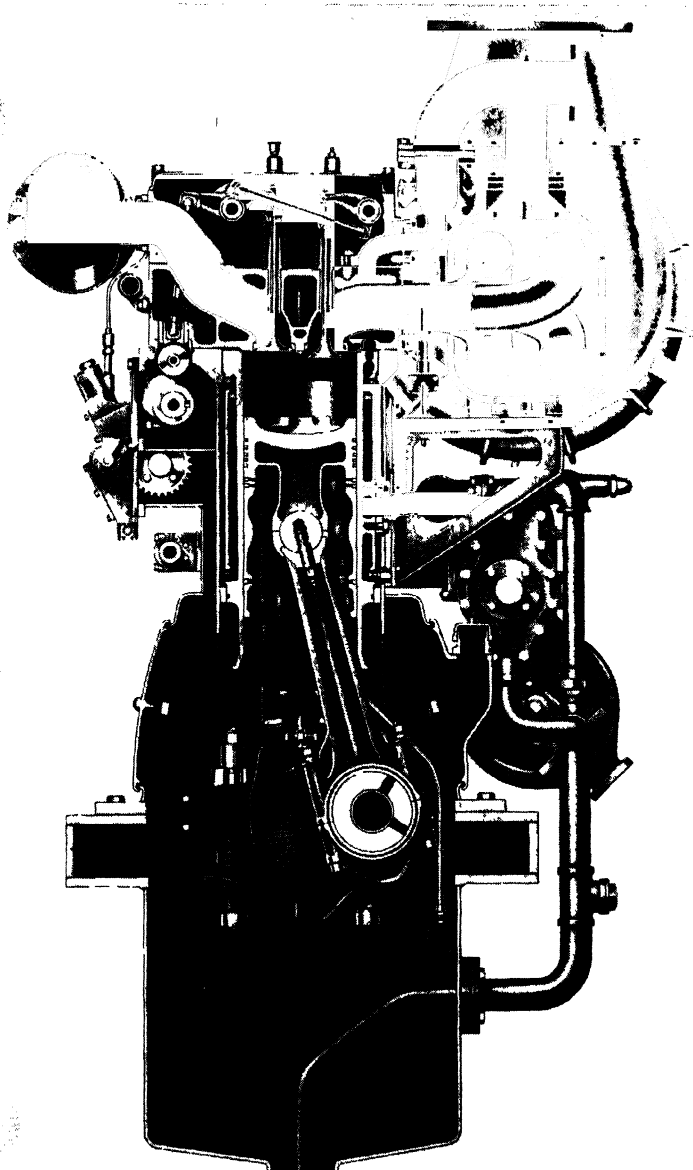


FIG. 10-9. Cooper-Bessemer engine.

casting the babbitt in the rapidly revolving box. Centrifugal casting produces better results; however, special equipment is required and the work can be done only in a shop having such equipment. When such a shop is available, sending the bearing to the shop is better than using the mandril method.

Pouring bearing metal. Mandrils are usually made from cast-iron stock, with a diameter about ⅛ in. less than the pin on which the bearing operates. The bearing boxes should be brushed with a solution of muriatic acid cut with zinc, or muriatic acid that has been dissolved with as much zinc as it will take. The bearing is heated on the outside by means of a torch while tin solder stick is rubbed over the surface to be babbitted, with some powdered zinc chloride being sprinkled ahead of the stick. The two halves of the box should be separated with shims of the same thickness as those used in operation and then clamped together. The box is then lowered over the mandril, which is standing upright, with the flange down. The mandril is centered and wet fire clay is molded around the outside of the box where it rests on the flange of the mandril. A high grade of babbitt must be selected to secure the kind of bearing required. Babbitt should be heated in the ladle by means of several blow torches. A babbitt is ready to pour when it will char a pine stick that is immersed in it. After the babbitt is cool, it should be bored to its final diameter, keeping it a little undersize rather than oversize. The halves are sawed apart, the edges are faced and cleaned, and the bearing is ready to be fitted to the pin.

Method of scraping bearings. It is never advisable to try to scrape a bearing to 100 per cent contact with the pin, nor would this 100 per cent contact be advisable. A bearing is satisfactory if the contact is about 75 per cent of its area. The bearing is placed on the pin after the coating of *Prussian blue* has been applied to the pin. It is then rotated through a small angle back and forth several times. When it is removed, the high spots can be seen by means of the bluing adhering to them. These high spots shown by the coat of bluing are scraped down. The procedure is then repeated. When high spots are scraped too much, a low spot will result, requiring the entire surrounding area to be scraped to it. For this reason, too much scraping should not be done in one operation. When the Prussian blue is applied too thickly, the bearing will be well coated and will give a false appearance of a good fit before the fit has been attained.

Precision-type bearings should not be scraped. Solid bronze bushings are sometimes not scraped but it is a good practice to try them out for bearing area and scrape if necessary.

Crankpin bearings for large engines are usually scraped to an accurate fit, but for some engines they are now bored very accurately to the proper size without scraping.

When bearings are bored to the proper size, the crankpin box should be assembled and tightened by the crankpin bolts, the same tension being used as that applied when in service.

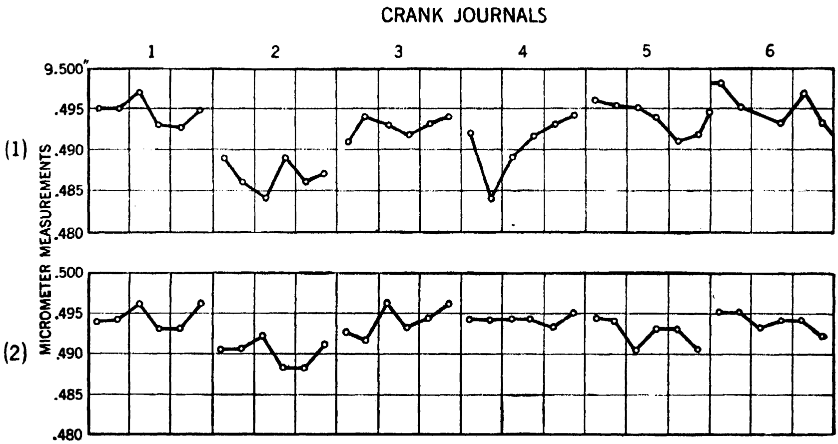


FIG. 10-10. Micrometer measurements plotted on a graph to show the results obtained by two different mechanics. Measurements are made across and with the throw, at each side and in the center of each crankpin and main bearing.

Otherwise, the difference in tension will distort the boring, and cracking of the babbitt is likely to result.

Cracking of babbitt bearings. Babbitt may crack for other reasons. If the bearing cap is not sufficiently rigid and distorts under load, the babbitt is certain to crack eventually. Sometimes, rebabbitting is avoided by the cracked portions being cleaned away and the surface being brought up with solder. A bearing that has run hot and dragged or wiped the babbitt may be scraped to a good surface if the thickness of the babbitt is not appreciably reduced.

The Magnolia Metal Company, producers of high-grade babbitt, furnishes a manual of instructions that is complete and reliable, and when possible, the operator will profit by using this information.

Causes of bearing wear and failures: check list. When reports on bearing wear or failures are made up, the records should include the information on the causes of the wear. The following summary is a guide and a check list of known causes of bearing wear and bearing failures:

1. Defective bearing construction, or assembly:
   a. babbitt too soft, or improper bond;
   b. babbitt too hard or brittle;
   c. shaft distortion.
2. Shaft distortion, which is usually due to:
   a. overspeeding;
   b. excessive bearing clearance;
   c. misalignment.
3. Overloading the bearing, excessive peak pressures, due to:
   a. too much injection advance;
   b. low cetane fuel;
   c. unbalanced cylinders.
4. Improper lubricating oil, or improper lubrication, due to:
   a. failure to use hand pump and lubricate when starting;
   b. improper grade of lubricating oil;
   c. corrosive substances or acids in the oil, water, sulfur from the fuel, and so on.
5. Lack of lubricating oil, due to:
   a. excessive bearing clearances;
   b. low oil level;
   c. low oil pressure;
   d. plugged oil passages or ducts.
6. Low temperature of the lubricating oil, due to:
   a. too much water circulation.
7. High temperature of the lubricating oil.

The influence of these factors on wear of bearings and other engine parts is more fully discussed in the chapter on lubricating problems.

Practical experience on the job affords the only opportunity for complete understanding of bearing maintenance. Papers reporting the experimental results of numerous tests should be consulted by students and instructors.

QUESTIONS

1. What are the characteristics and properties of a good bearing metal?

2. What kind of metal is used for high-speed engine bearings?

3. What causes of bearing wear are less apparent than others?

4. How does abnormally high bearing pressures affect wear?

5. What are the chief factors that contribute to the wear on the main bearings?

6. What may be expected when the engine is operated with excessive bearing clearances?

7. What factors are responsible for vibration of the shaft, and what effect does this have on the main bearings?

8. Why is wear found on the upper main bearing shell when the weight of the rod and piston and the force of the explosion are always downward?

9. After a fatigue failure is repaired, what inspections should be made after the bearing is put back into service?

10. Why do foreign particles in the lubricating oil cause bearing failures?

11. What is bearing wiping?

12. What causes maximum, or peak, pressures to be too high?

13. What effect do the corrosive properties of oil have on bearings?

14. What causes vibration when the engines are connected to driven machines?

15. Can cracked babbitt shells be repaired efficiently, and put back in service without replacing the lining?

16. Can a cracked babbitt shell be corrected by welding?

17. Do you know what clearances and what wear limits for the bearings of your engine should be?

18. Name three examples of faulty assembly.

19. What may occur when waste instead of cloth is used for cleaning engine parts?

20. How are oil ducts cleaned when high-pressure air or steam is not available?

21. When bearings are being fitted, should a torque wrench be used to obtain proper alignment?

22. Do burrs on the connecting rod bolts prevent proper tightening when a torque wrench is used?

23. Why is it important to make sure that oil holes in the bearings coincide with the oil holes in the bores?

24. Does reversing the upper and lower halves of the bearing have a destructive effect?

25. Can connecting rod misalignment cause connecting rod bearing failures?

26. Overloading is said to cause failure of the top connecting rod and lower main bearing failure. Will overspeeding cause just the reverse?

27. Why is it that the inertia loads on an overspeeding engine are greater than the combustion loads?

28. How does starting the engine repeatedly under unfavorable conditions, such as cold weather, produce heavy shock loads that deform the bearing metal?

29. Would you look for dirt particles behind a failed bearing?

30. Are high firing pressures the result of by advanced injection timing?

31. Will this high firing pressure cause heavy bearing loads? Would the use of a fuel with a low cetane rating have some relation to this condition?

32. Does fuel dilution, high oil temperature, or the use of an improper grade of lubricating oil of insufficient viscosity, cause low lubricating oil pressure?

33. Why does lead segregation in the copper-lead bearings, or too much lead in the tin-base babbitt, cause a bearing problem?

34. Bearing failures due to localized overheating are usually caused by the lack of conformability and poor embeddability. If failure occurs for this reason, would you say that the bearing was the proper one for the engine?

35. What kind of bearing failure results from repeated application of heavy loads which eventually cause cracking of the metal?

36. Failure due to poor fit is caused by the load being concentrated. What result is expected?

37. What kind of failure results when continued use of oxidized oil is permitted?

38. When oxidation products are formed, what effect do they have on the lead or cadmium?

39. What causes pitting of bearings?

40. Lead removal from the copper-lead bearing is due to "sweating out" of the lead. What causes the "sweating out"?

CHAPTER 11

PROBLEMS OF LUBRICATION

Introduction. The mechanical methods of applying the lubricant to the various parts of the engine have been developed to a point where the effective lubrication of the average Diesel engine is almost foolproof. The modern high-pressure lubricating systems have reduced crankshaft and bearing wear to a small figure compared with that of some years ago. Provided a film of lubricating oil is maintained between working surfaces, it is not likely that actual failures often occur, as the oil is directly applied to bearings and all working surfaces through ducts and pipes under pump pressure and must reach all points to be lubricated. Most of the problems of lubrication are concerned with the oil itself, and what happens to the oil when used in the engine.

Contamination of the lubricant. During the operation of an engine, certain heavy ends of the fuel oil find their way into the crankcase and dilute the lubricant. Some of the burned products get into the crankcase by working past the rings. In addition to these, in the lubricating oil will be water, dust, bearing cuttings, and free carbon. In other words, the oil originally supplied becomes contaminated. When an improper or inferior grade of oil is used, it is almost certain that the body, or the viscosity of the oil, will drop below the point of safety, and damage usually results. The contact of the lubricating oil with the high heat of the combustion flame would destroy the film of lubricating oil on the cylinder wall if the cylinder were not properly cooled.

If the jackets are filled with water at all times, however, experiment has shown that it is almost impossible to burn the oil film that is on the cylinder wall. The cylinder wall oil film is very thin, and it must be at a temperature approximately that of the wall itself, which is in turn about the temperature

of the water in the jackets. It is evident that any oil with a flash and a fire point of 400° F should stand up and not crack or burn while the cylinder wall is in contact with the combustion flame for the brief intervals during the firing period.

The percentage of good lubricating oil that is burned or cracked by such means is very small and usually the burning of the film by contact with the flame is indeed small and harmless. Properly operated, no other part of the engine attains a temperature sufficient to crack the oil. Any oil that reaches the combustion chamber by passing the rings will burn as fuel and will form carbon deposits. The contact of the oil with hot spots under the piston causes overheating and cracking of portions of the oil if the engine runs unduly hot. The contact with the hot surfaces causes decomposition of the light fractions, which evaporate and leave the crankcase by the way of the breather pipe or form into carbon deposits. This carbon formation mixes with the lubricant and gives it a dark color. The remaining oil, plus the contamination, makes up what is left in the crankcase. These parts of the oil that cracked against the hot central portion of the interior of the piston do not remain in the crankcase as oil. If these contaminants are generally and regularly removed from the oil and the remainder treated, the oil will be practically as good as the original oil. It is the purpose of lubricating oil filters to take care of the contamination.

When a lubricating oil is properly filtered or reclaimed, it should have a viscosity higher than that of the original oil. This is due to the fact that the oil has had removed from it those light hydrocarbons that are not removed at the refinery. The oil in the engine actually undergoes further distillation during its operation in the engine, and during the filtering and reclaiming process. Modern methods of filtering and reclaiming oils restore practically every property the oil had when new, both physical and chemical; and since it has been further refined, it is actually superior to the original oil, with a higher viscosity. It is quite possible that it is superior because it will certainly be free from those molecules that cracked against the hot pistons. It should also be comparatively free of the undesirable unsaturated hydrocarbons left in the new oil by the refinery. When it is again used in the engine, the oil should deposit less carbon than it did when used the first time in the engine, since it is free of the easily cracking molecules. It is those molecules in lubricating oils known as *unsaturated hydrocarbons* that cannot

stand the high heat of the engine during the operation that constitutes the basis of many lubricating problems. These light hydrocarbons are easily broken down during the normal operation of the engine. Removing these unsaturated hydrocarbons in the original refining process would be too expensive. No refinery could spare the expense to refine an oil as the engine itself can do during its continuous operation.

How sludge is formed. When the oil is heated to the engine operating temperature, and at the same time, agitated with air, a definite oxidation occurs. The oil darkens in color and sludge begins to form. This sludge is recognized as a slimy, thick, viscous substance, which leaves deposits in the lubricating system. There is a maximum amount of sludge that will form from new oil in this manner, and after that maximum is reached, continued agitation will not form more. The operation of an engine comprises a perfect sludge-forming process. When the sludge-forming hydrocarbons have formed into actual sludge, the sludge is removed by filtering. This is why the oil becomes better after it has been used for a certain length of time, for it has had the original sludge-forming hydrocarbons filtered out of it.

It should be understood that all mineral oils become more or less oxidized during the agitation in the presence of high temperature and water together with water vapor in the engine crankcase. Where particles of air and water are suspended or retained within the oil itself, and where there are light hydrocarbons in the oil, oxidation is a natural chemical process that must occur as a result of the use of the oil in the engine under these operating conditions. This applies to even the best of oils. The reclaiming of the oil removes the acids due to this oxidation. It is the process of refining that renders lubricating oils as stable as possible, but it is practically impossible to render an oil absolutely nonreactive when it is subjected to detrimental conditions in the presence of air and water under high temperature as well as dust and dirt that come in with the intake air charge. This foreign matter actually forms a major part of certain kinds of deposits found in the engine. In addition to the abrasion and corrosion, such foreign matter acts in the same manner as the catalytic action of the metallic particles.

The maintenance man and the operator require a considerable amount of technical and related information for a

thorough understanding of these problems of lubrication. It has been found that the bulletins and technical publications distributed by the major oil companies should be obtained and studied by the operators of Diesel engines. The refining and manufacture of lubricating oils is a highly technical process, highly developed and scientifically controlled by experts. It is evident that the average operator would require a very specialized training in petroleum technology to select and use the proper grade of lubricating oil on his own judgment. He therefore depends upon and must place a great deal of confidence in the reliability and reputation of his supplier of oil. The reliability of the refinery is the most important consideration in purchasing oil for use in a Diesel engine. Here can be set forth only the kind and nature of problems of lubrication, and how the operator may determine by experience and observation when he has problems requiring the attention of expert lubricating engineers.

The continuous oxidation of the lubricating oil requires attention constantly. When it is continuously agitated in the presence of water vapor and at a high temperature, oxidation occurs. The amount of this oxidation depends upon the balance of operating conditions rather than on the grade of oil itself. And only a slight amount of air and water mixed with the oil with the necessary temperature and agitation is needed to start the oxidation reaction between the air and the water. The extent to which the oxidation occurs depends upon the degree of refinement of the original oil and a number of other factors. Certain hydrocarbons oxidize more than others. Unless the refinery has removed them through a very thorough and correct refining process, the oil will oxidize more readily and may not be as satisfactory as it should be. It is here that the reliability of the refinery is the controlling factor.

Factors that promote oxidation. Oxidation is said to be accelerated by the chemical effects of various metals, such as brass, bronze and iron, and perhaps by various kinds of foreign matter. It has been found that the presence of foreign particles of dust and dirt in an already emulsified oil promotes oxidation in forming insoluble sludges that are very harmful to lubrication. For this reason, many engineers say it is important to prevent, or reduce, emulsions of oil and water as far as possible. It is believed that if emulsion can be prevented, sludging will be, for all practical purposes, prevented in the engine lubricating

oil. It should be remembered that emulsion is simply oil and water agitated with the oil held in suspension.

It is likely that the "catalyzer," such as iron particles in the sludge, really promotes the oxidation. The emulsions themselves are not likely to clog the oil passages. It is the rust that first begins to form around the particles of foreign matter, and it is this rust that forms sludge, restricts oil ducts, and impairs the lubricating efficiency of the oil.

The quality of the lubricating oil is restored by filtering and on many installations by the use of the centrifuge. Filtering systems are designed to remove the water, foreign matter, and sludge as rapidly as these are accumated in the oil.

Nature of oxidation process. It has been said that sludge formed by agitation of oil and water in the presence of air and high temperature goes through two steps: (1) The colloidal or soluble stage is the initial development. (2) The sludge thus formed then becomes insoluble or permanent in form. The colloidal stage occurs when a stable emulsion is formed by the presence of water; however, colloids may not be present at the time the emulsion starts to form. Such emulsions and colloidal sludges usually disappear at normal operating temperatures when the emulsions are not contaminated with the foreign matter, or aided by high temperature in oxidizing, and these are cleared up by precipitating in the water. Sludge should be settled out when standing, with the result that the oil should be in the same condition as before agitation incident to the operation.

It should be kept in mind that colloidal sludges are detrimental even when they can be clarified. They interfere with the formation of a continuous film of lubricating oil over the surface that is being lubricated. Colloidal sludges should be removed as they are formed; if not, they are absorbed by the lubricating oil at the operating temperature. There are good and sufficient reasons for removing these sludges as they form in the oil. When left in solution with the oil, the sludges help break down the oil and this interferes with the separation of the contaminating foreign matter, which in turn, exercises a catalyzing effect that promotes further oxidation; this leads to the formation of the permanent sludges that cause most of the lubricating oil problems. Removal by clarifying or centrifuging has for many years been the general practice in dealing with this type of sludge formation in all large plants.

Crankcase deposits. An analysis of crankcase deposits indicates that several kinds of substances are found in the average engine. These sludge deposits fall into two or three classes, depending on the nature of the sludge and how it is formed. The sludge formed by oxidation has been described. In the crankcase deposits is also found a sludge having the characteristics of tar. This tar forms as a result of the combination of oil with oxygen at high temperature. Oxidation products of this kind are formed directly in the oil, and are not due to contamination as are other types of sludge. Still another kind of sludge is formed when the carbon originating in the combustion chamber, identified as soot, unburnt lubricating oil, and fuel oil carbon, works down into the crankcase. This sludge is composed of carbonaceous material not completely oxidized, or burned, and hence is in the form of soot. Such carbonaceous deposits also coat the combustion chamber and the top of the piston. When such carbon deposits work past the piston rings, and into the crankcase, they mix with the lubricating oil, and are carried to various parts of the engine; this results in clogged oil ducts and strainers. This kind of sludge is black in color, and is found to comprise a great deal of the deposits found in the ring belt.

The presence of soot due to incomplete combustion indicates that the fuel injectors should be checked, cylinders balanced for load, and generally, the injection timing should be checked. When the time for the renewal of the pistons, rings, and liners approaches, more and more evidence of incomplete combustion is noticed. The engine fouls up sooner and more cleaning is required to keep the rings working. The continuation of this kind of deposit and sludge also suggests the need for checking the fuel quality; and it perhaps indicates that a higher cetane fuel may be advisable. The use of detergent oils reduces this sort of sludge formation in the high-speed engines. Frequent draining of the oil, and proper attention to all filters tend to keep the amount of this kind of sludge to a minimum consistent with safety and longer life of the engine parts.

Oil tar sludge. The tarry kind of sludge, black and sticky, is attributed generally to the prevailing high operating temperatures. Engines operating at high temperatures, especially when the lubricating oil temperatures are above the recommended level, usually produce a considerable amount of tar. This tarlike substance in sludge is usually found on the piston

skirt, where it is indicated by its brownish color, although it is often black when soot is mixed with it.

A sludging study has been made of various kind of oils. The solution to the problem of sludge in a Diesel engine at the present time comprises the use of detergent oils in practically all high-speed engines. This oil is usually referred to as "additive," or "compounded," or "detergent." These oils consist of the base mineral oils to which has been added the compounding material. The usual additive agent has certain beneficial effects on the performance of the basic lubricant. These effects are as follows: (a) It acts as an inhibitor; (b) it improves the natural detergent property of the oil, namely, the ability of the oil to remove or prevent the formation of carbon deposits; and (c) it increases the affinity of the oil for the metal surfaces.

Advantages of detergent oils. The use of the compounded oils for the lubrication of a Diesel engine helps reduce ring sticking, and prevents gum or varnish formation on the piston and such parts. When used in dirty engines, the detergent oil gradually softens and helps to remove the gummy, carbonaceous deposits. This material is removed from the engine surfaces, but it is then carried in suspension in the oil, and, if too much is accumulated without change of oil, it may clog the oil filters. When the oil is used to clean a dirty engine, the crankcase should be drained two or three times at frequent or short intervals until the engine is thoroughly purged of deposits. During the cleaning-up process, the operator should drain the sump and clean the filters at any time that the gauge indicates an inadequate flow of oil.

Drain periods. Compounded oils contain various percentages of the compounding materials, and the amount of such additives determine its efficiency or life. The material added to the oil is consumed in preventing the formation of sludge and deposits. It is for this reason that the oil must be drained and replenished at regular intervals. Oil drain periods, according to the experience of many authorities, should be governed by certain limits in order to obtain the most efficient results from the performance of the oil. When a chemical analysis shows that the following contamination has been reached, the oil should be drained: (a) Neutralization number, 0.5 maximum; (b) precipitation number, 0.5 maximum; and (c) fuel dilution, 5.0 per cent maximum. These limits apply to straight-run mineral oil as well as to the compounded oils. It is considered good

practice to govern drain periods accordingly, although some margin of safe operation lies beyond these limits. This results in drain periods for large, slow-speed engines of about 500-hr duration, whereas for small engines, of the high-speed and medium-speed types, draining every 100 hr is recommended.

Precautions with additive oils. Additive oils should not, as a rule, be mixed with straight-run mineral oils. It may be done in emergencies, but the practice is not recommended. The use of the earth type or the chemically active filters with additive oils is not recommended, as this type of filter removes the additive. The most acceptable type of filters for use with additive oils are the cotton waste, yarn, or cellulose type. It is not possible to judge the performance of the filters by the color of the oil, as detergent oils become dark in color almost immediately, as a result of the fine particles of carbon suspended in them, which, however, are not harmful. It is the purpose of the additive to dissolve, remove and hold in suspension the carbon in the form of particles too fine to cause trouble. The filters catch some of this suspended carbon. For this reason, the filters should be changed and cleaned at the same time that the oil is changed. The use of noncorrosive, additive oil is usually specified; however, very few of the additive oils can be said to be corrosive. When surfaces of the metal are etched, or the bearings are corroded, it is usually more logical to assume that the contamination of the lubricant by water and partly burned fuel oil is responsible, and not the lubricating oil itself. It is for this reason that the fuel system should be kept in good repair and adjustment at all times. Other precautions are to see to it that water, partly burned fuel, and all foreign matter are removed from the oil in the crankcase. Difficulties, too often blamed on the lubricant, are, as a matter of fact, due to poor operating practice, the design of the engine itself, or to the kind of fuel used.

Foaming and its causes. There is a tendency for additive oils to foam when the quantity of oil used is large. All oils foam when aerated; however, foam produced in the compounded oils is more tenaceous and persists longer than that produced by the straight-run mineral oils. The aeration of mineral oils without additives is more serious from a service standpoint than the aeration of compounded oils, as the straight-run mineral oils quickly oxidize, form sludge, and develop acids, while the compounded oils contain the oxidation inhibitors that

reduce the rate of oxidation of the oils. It is evident that it is important to prevent, as far as possible, the aeration of the lubricating oil of any kind.

When there are air leaks in the suction side of the oil pump, the oil may be so aerated that it will form foam. Foaming due to such leaks can be eliminated by repairing the leaks. The discharge oil lines should bring the oil back to the sump or into the tank at and above the oil level, thus allowing the entrained air to escape. A device installed to permit the air to escape from the oil is another precaution against foaming. Foaming is usually due to some mechanical operation, such as pumping, and involves the function of the lubricating system rather than the oil itself, or the operating condition.

Centrifugal purification. The centrifuge can remove most of the dirt, mineral matter, and water and much of the carbon. It does not remove soluble oxidation products and fuel dilutions. The centrifuge can keep the solids down to a low level; however, the oxidation products as indicated by the neutralization number increase, as does the viscosity. Lighter oils are used as makeup to prevent this increase in the viscosity. Sometimes, stable emulsions are formed in heavy-duty oils and cannot be removed by ordinary centrifuge operation; it is possible, however, to heat the oil at the centrifuge to 180 to 200° F so that the emulsions can be readily removed by centrifuging.

The engine builder's instruction book should contain very detailed instructions relative to the routine required to maintain the proper lubrication of the engine. A strict adherence to this routine is the first essential duty of the operator and maintenance man.

Filters for lubricating oils. A wide variety of filtering devices is available. Some of these filters on the market may remove very little of the contaminant from the oil, whereas some of them take out everything except the oxidation products, water, and fuel dilution. Standard fuel and lubricating oil filters have been developed in great numbers. There are efficient filters for lubricating oil of the detergent type. However, the advice of the engine builder should govern all decisions involving the selection of filters and mediums used for filtering detergent oils.

Recent developments. In a paper, entitled "Modern Filter Developments as Applied to Fuel and Lube Oil Systems of Diesel Engines," by C. A. Winslow, read at the Northern

California Section of the SAE, this subject was discussed in some detail. Mr. Winslow detailed certain points now good practice in the filter field. A portion of his paper is abstracted here. The primary purpose of fuel filters is to prevent grit, gum, abrasives, varnish, and other impurities from passing from the fuel oil supply to the delicate and closely fitted working surfaces of the injection pump and still more delicate parts of the fuel nozzles, he says. Because the filter must be a one-pass filter, it is essential that the porosity through the filter be of such nature that the minimum dangerous particle size will not pass through the filter under the maximum velocity and pressure that can be developed by the supply pump. An ideal filter hookup would be one where only the fuel actually used by the engine passes through the filter. The best precaution that any Diesel engine operator can take to insure continuous operation with clean fuel is to provide adequate settling space before the fuel is delivered to the supply tank, and to insure positive filtering through adequately large filters at the time the fuel is delivered from the storage to the supply lines.

Regarding Diesel lubricating oil filters, the magazine *Lubrication*, published by the Texas Company, states: "Precision bearings and cap and saddle bores into which they fit are machined to very close tolerances. With bearing linings of 0.002 to 0.005 in. thick, and with journal bearing clearances of 0.0015 to 0.0035 in., the matter of a few thousandths of an inch becomes important. The life of a bearing in service will depend on how well these clearances can be maintained. Thus during fitting or bearing replacement, a misalignment of the order of 0.0010 to 0.0005 in., or a reduction in clearances of the same magnitude, may prove serious. Dirt is perhaps the greatest enemy of bearings. Engine manufacturers install filters to keep dirt out of engines because they realize that the life of engines can be materially increased if the dirt is kept out."

Since there are so many kinds of lubricating systems to which filters are attached, or of which filters are a part, the accompanying illustrations are given to show some features that are applied with variations to many Diesel engine installations. Fig. 11-1 shows a simple shunt-type filter installation wherein the filter merely steals a certain amount of oil from the gauge line, filters it, and discharges it back into the lubricating oil supply. This is the conventional system used on most automotive and

vehicle engines that are not designed to receive the filter as a built-in part of the lubricating system.

Fig. 11-2 shows a full-flow in-line system in which the filter is connected directly between the pump and the engine bearings. A by-pass is included in the filter that insures that the oil will be

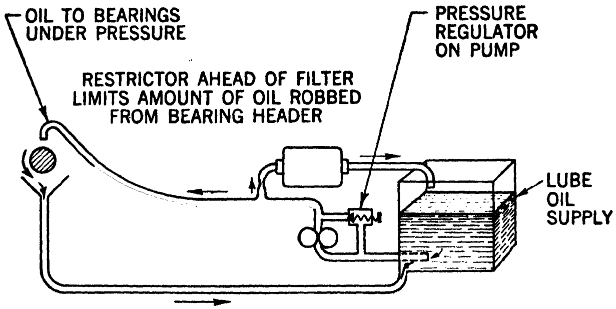


Fig. 11-1. Conventional automatic-type system—one pump. This system affords bearing protection only if oil is maintained clean. Any grit in the lubricating system is pumped directly to the engine bearings.

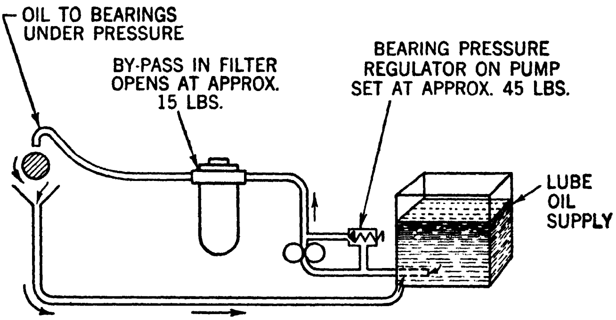


Fig. 11-2. Full-flow, in-line system—one pump. This system provides simplicity of design with maximum bearing protection and adequately large filtering area.

delivered regardless of the function of the filter. Fig. 11-3 shows a conventional pressure system in which the primary pump forces the pressure against the engine bearings, with the by-pass discharge back to the oil supply. This oil filter is operated by a separate pump while the pressure regulator from the intake to the discharge limits the maximum pressure against the oil-filter element. No by-pass is needed on the filter in this case, as it stops when the elements are completely clogged

up, and has nothing to do with the delivery of the oil to the engine bearings.

Fig. 11-4 is a modification of Fig. 11-3, except that it is generally used with the dry-sump engines, the similarity being that two pumps are required, one to insure the pressure against

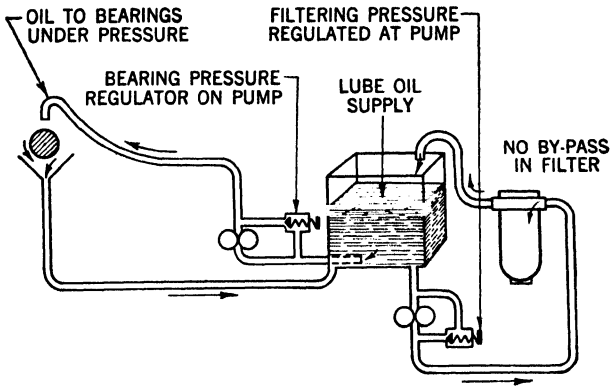


FIG. 11-3. Shunt-type system—two pumps. This system is similar to the one shown in Fig. 11-1, in that the bearings are protected only as long as oil is maintained clean. Grit or metal can be pumped directly from the crankcase to the bearings.

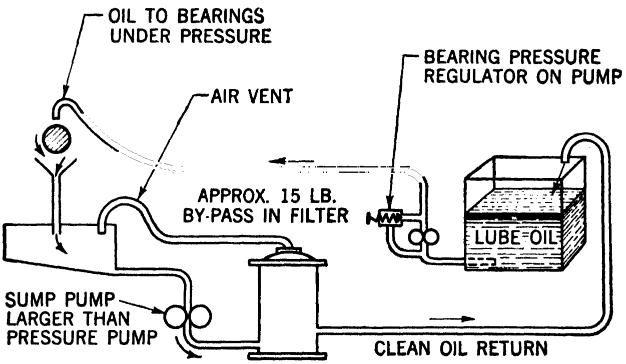


FIG. 11-4. Dry-sump system—two pumps. This system affords maximum bearing protection and is recommended for all large installations.

the engine bearings, and the second to scavenge the crankcase, to force the oil, hot and dirty, from the sump, through the filter to the oil supply tank.

It is evident that not only must oil filters be changed and cleaned when necessary, but that also careful consideration

must be given to the type, size of filter and its application. The engine builder's engineers are doing constant research to determine the best filters and recommend the proper practice in operation and maintenance of filters.

The manufacturers of Diesel engines, as well as the oil companies, research men and others, have given the utmost consideration to the selection of lubricating oils for the engines of today. Several important requirements are involved in the selection and proper use of lubricating oils: (a) minimum maintenance requirements, (b) improved operating efficiency, and (c) lower cost of operation. The requirements of Diesel lubricating oils, generally speaking, are that they should (a) lubricate all engine parts and bearings satisfactorily, (b) be noncorrosive to bearings and other parts, (c) prevent ring sticking and oil ducts and passages clogging, (d) reduce cylinder and ring wear to a satisfactory minimum rate, (e) prevent excessive carbon deposits on any part of the engine, and (f) be economical to use. Any lubricating oil that satisfies this bill of requirements is acceptable.

Organic acidity. The products of oxidation of an oil are acidic. The neutralization number is a measure of this organic acidity. These acids are harmless in the ordinary sense, but they cause the oil to dissolve the lead or cadmium in the bearing composition. Thus, the increase in the neutralization number of a clarified oil measures the amount of soluble oxidation products present that may form varnish or lacquer. The neutralization number of some kinds of detergent oils gives a false result, with consequent fictitious estimates, and such a test for these oils is not applicable.

The acidity measured by the neutralization number is defined by the American Society for Testing Materials as the *weight in milligrams of potassium hydroxide required to neutralize one gram of oil.* This test is considered an important one in the study of Diesel engine lubricating oils. A maximum acid, or neutralization number, has been agreed upon by certain authorities as a safe limit for Diesel crankcase oils, the tests being made at periodic intervals, depending upon the kind of oil in question and the service to which it may be subjected.

Another means of testing the corrosive properties of a straight-run mineral oil is the use of the so-called copper-strip method. The strip is immersed in the oil for three hours at 212° F. If the oil corrodes the strip, it may also be corrosive to

steel and other engine metals. It is a characteristic of many additive oils to give a dark, tarnished color to copper-lead bearings. This does not indicate corrosion of the bearings. A corrosion test is made to determine when the oil becomes corrosive. The corrosive property of the oil may be determined when it comes from the crankcase by means of the copper strip test, which is very good for this purpose.

The principal value of the analysis of used oil is the measurement of the amount of contaminants, which include foreign matter, mineral particles, oxidation products, carbonaceous matter, water and fuel dilution. The tests that show the condition of the oil itself are neutralization number, copper-strip corrosion, bearing corrosion, and additive content.

Routine problems of lubrication. There are a number of routine problems of lubrication for which the instruction book sets up a procedure of operation and maintenance. Some of these are discussed here to show the relation to other engine and operation problems.

1. Use of excessive lubricating oil. Investigation of all leaks and other disarrangements bearing on excessive lubricating oil consumption should be done promptly. Leaky rings, worn cylinders and worn bearings are on this list of things contributing to excessive lubricating oil.

2. Low lubricating oil pressure. Broken oil lines or leaky connections may be involved. Oil pump gears may be worn; even worn or loose bearings may result in low lubricating oil pressure. A dirty or stopped-up oil filter may cause low oil pressure. Check all of these when it is first noticed that the oil is excessive.

3. Dilution of the lubricating oil. The same factors that may result in low oil pressure, also permit the dilution of the oil in the crankcase. Leaky injection nozzles, dribbling nozzles, and the like, including worn injector plungers, worn fuel pumps and scored plungers may have some relation to this problem.

4. Influence of idling on dilution and sludge formation. It is important to avoid idling of the engine for long periods when this results in inevitable dilution of the lubricating oil. The prevention or removal of the sludge has been discussed.

5. Prevention of dilution. The engine should not be operated with overloaded cylinders or run with smoky exhaust and with compression leaking past the piston rings. Smoke at the exhaust may be due to various causes other than overloading,

but the cause of the smoke should be determined by careful investigation.

FIG. 11-5. Lisle magnetic plug used instead of ordinary drain plug for crankcase or lube oil sump. This plug has anchored in it a magnet which attracts abrasive metal particles resulting from wear of bearings and gears. It is shown at the left before operation and at the right after operation.

6. *Periodic check.* Several parts of the engine must be kept in good condition for operation if dilution is to be avoided. The piston rings must function properly; stuck rings must be prevented. The valves must not leak, and the fuel injection system must be kept correctly adjusted and timed. Careful inspection of all filters, both fuel and lubricating oil, helps to insure proper lubrication. It is also very important to maintain the air filters in good condition and keep dirt and grit out of the engine. The engine should not be operated without an air filter, because crankcase sludge and formation of carbon are due in part to the dirt that comes in with the air, as has been shown.

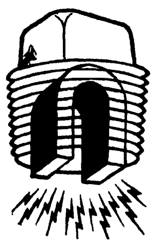


FIG. 11-6. Phantom view of magnetic plug, showing location of magnet in regularly constructed plug.

7. *Starting procedure.* The correct procedure in starting the engine should be observed. In cold weather frequent starting and stopping of the engine aggravates the

lubricating troubles. It is evident that a good deal of fuel dilution occurs when starting an engine even under the best conditions.

8. Pressure adjustment. Improper adjustment of the pressure-regulating device, failure to maintain correct oil pressure, and failure to investigate the cause of low oil pressure are the chief sources of many problems of lubrication. A routine inspection is usually detailed in the instruction book, and should be carefully followed up and complied with in every detail. The following should be checked before actually making any adjustments of the oil control or pressure relief valve:

a. Determine if the oil is too thin for proper lubrication.
b. Check the oil supply to determine if it is low or the system is leaking.
c. Determine if the oil filter is clogged, or the filter elements fouled up.
d. See that the oil screen in the crankcase is not clogged, as a result of sludge, dirt, and foreign matter.
e. Examine all oil lines for leaks.

When the low oil pressure cannot be traced to any of these, the pressure relief or oil control valve may need adjustment. Clogging oil lines, a common enough thing in engine maintenance, can be prevented through inspection and care in handling the oil supply, taking the necessary precautions to keep it clean, and prompt maintenance of filters that are dirty as a result of contamination. The entire filter should be dismantled and cleaned at intervals, on general principles. When screens clog, the flow of oil is retarded, and a reduced quantity is pumped through the system. The screens, therefore, must be kept clean and free of gummy deposits, foreign matter, and any lint that may form part of the obstruction to the free flow of the oil through them.

9. Exhaust valve lubrication. The lubrication of the exhaust valves of the engine is a special item. A very small amount of oil is required, but the value must be lubricated. On the other hand, when excessive oil is used, the valve is corroded and carbon is formed on account of the high heat that bakes the oil on the valve and stem. These several minor points in connection with the lubrication and maintenance of the engine should be thoroughly appreciated. When the exhaust valves are sticking, they may be freed with a mixture of kerosene and

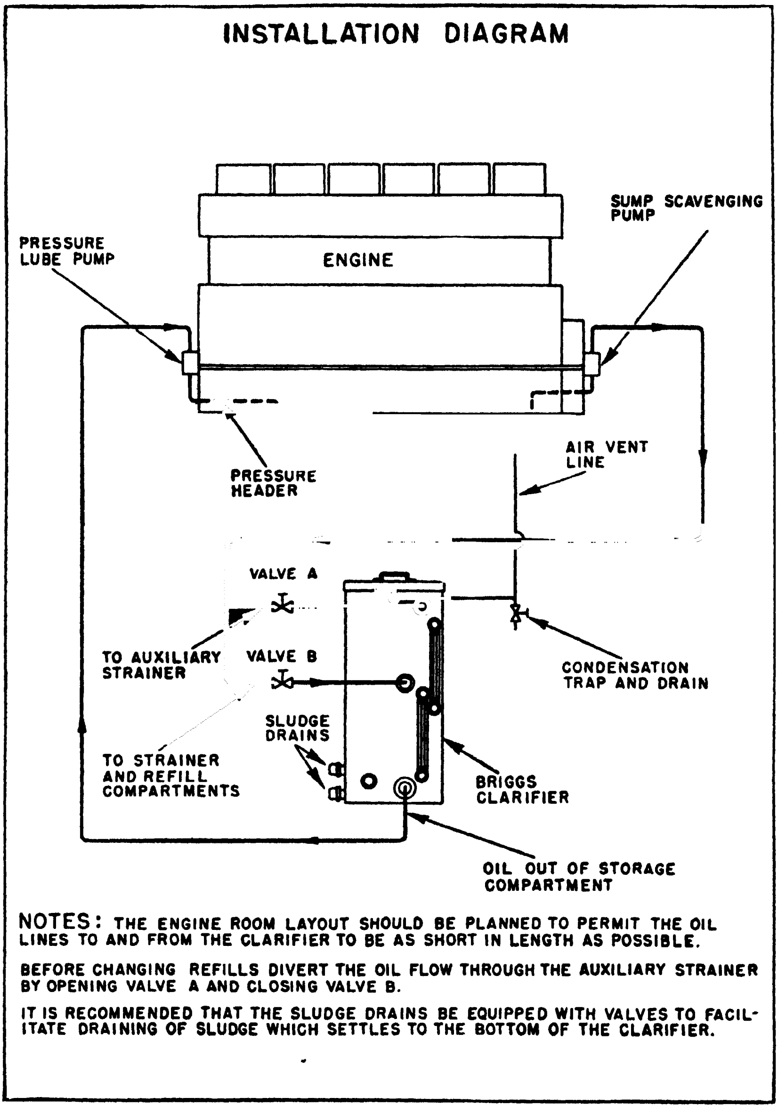


FIG. 11-7. Layout of installation of clarifier (Briggs) showing general piping arrangement.

lubricating oil applied to the valve stem. Idling the engine sometimes starts the exhaust valves sticking.

10. Miscellaneous items. The color of the oil has practically nothing to do with its lubricating value. Good oils should be

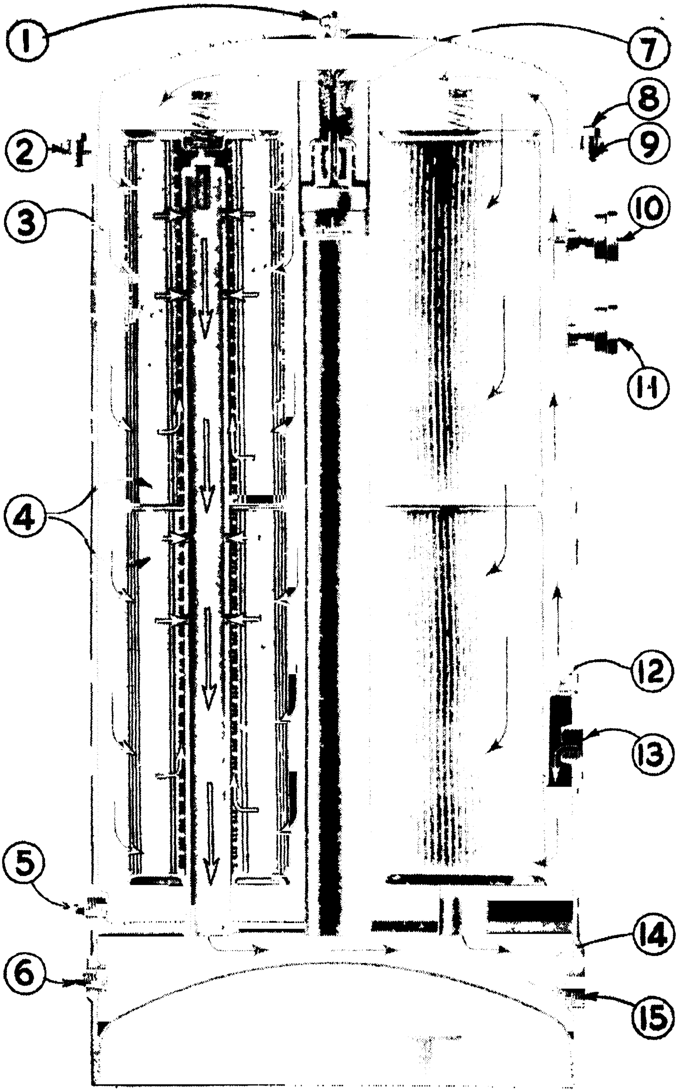


FIG. 11-8 The Briggs oil clarifier operates to maintain the lubricating oil with the precipitation number .05 and neutralization number .3 by the use of Fuller's earth block refills, which absorb dirt, carbon, metal particles, and adsorb soluble acids, gums, and resins An all-cellulose refill is used with detergent oils, the clarifier being so constructed that the cellulose and Fuller's earth refills are interchangeable.

low in acidity, since acid sets up the corrosive processes. Important considerations are acidity, carbon residues, poor tests, flash and fire point, and demulsibility. The divisions of oils into light, medium, and heavy classes at atmospheric temperatures means very little. There is no standard by which this classification is determined; it is misleading, and should be abandoned in our thinking of oils. Oil does not wear out. Oil drained from the crankcase is just as high in quality as it ever was. It is contaminated, but the oil itself is uninjured by its use in the engine. When reclaimed and corrected for viscosity by addition of some bright stock, the oil is better, as a rule, than when it was first placed in the engine. The engine is a refiner, and it is only necessary for the filtering to remove the contaminants.

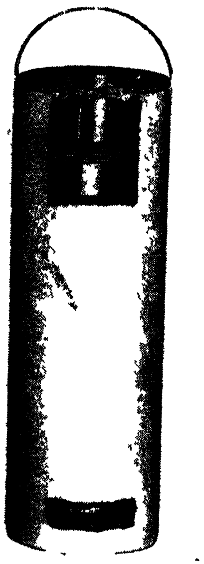


Fig. 11-9. Cellulose refill for Briggs filter. This type of filter refill is used for small Diesel and gas engine applications having pressure lubricating systems and hydraulic systems and using detergent oils.

Summary and check list. The causes and effects that involve problems of lubrication are factors that may be related to the mechanical adjustments, the lubricant itself, or the failure of other parts of the engine. The following comprise a check list which should be used for reference in dealing with the problems of engine lubrication:

1. High lubricating oil temperatures. There are various reasons for high lubricating oil temperatures, all of which should be understood, and checked by the operator whenever any evidence of such troubles or difficulties is encountered:

a. Insufficient oil in the oil sump, or pump failure.
b. Insufficient crankcase capacity for the engine load.
c. Insufficient or low oil circulation and pressure.
d. Clogged up oil cooler, or filters.
e. Incorrect oil viscosity, oil too thin, and dilution.
f. Sludge coating the crankcase, and preventing cooling.
g. Overheated bearings; pistons and rings running hot.
h. Insufficient jacket water cooling.

i. Dirt coating the outside of the crankcase, as occurs to vehicles that are operated in extremely dirty and dusty places.
j. Late burning of the fuel, injection retarded.

2. Abnormal crankcase deposits. Whenever the crankcase deposits are abnormal, as evidenced by high oil temperatures, and other troubles, the following may be related to the problems:

a. Water in the crankcase, causing sludge. This may be due to blow-by.
b. The cylinder cooling system and piston cooling may not be functioning properly. Check the entire cooling system.
c. The lubricating system may be fouled and heavy sludge formation taking place.
d. Mineral matter from dust, cylinder wear, bearing wear, and dust from the atmosphere.
e. Carbon from partly burned fuel oil, residue from evaporation of the oil film, oxidized oil particles.
f. Oxidized oil, due largely to blow-by, improper oil, excessive oil spray, and high oil temperatures from hot spots on the pistons.

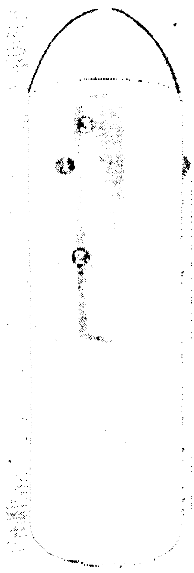


FIG. 11-10. Fuller's earth refill for Briggs filter or clarifier. This type of filter is usually designed for a by-pass installation off the oil pressure line, or installed independently with an auxiliary pump or pump and motor. It is used with straight-run mineral oils.

3. Excessive lubricating oil consumption. A number of factors are usually involved, but the causes are usually not far to seek:

a. Excess oil on the cylinder walls, due to overlubrication.
b. Bearing clearance too large, excess bearing end play.
c. Ineffective oil ring control. (Inspect the rings.)
d. Excessive piston clearance, worn liners, and pistons.
e. Cylinders, rings, and ring grooves worn.
f. Insufficient ring gap and stuck piston rings.
g. Scraper edge of oil rings worn off and rounded, or the oil rings clogged up.

h. Insufficient oil viscosity, due to new oil being too thin, or light, fuel oil dilution, or thinness due to high oil temperatures.
i. Oil leaks, wrist pin plates too tight, excessive suction on the crankcase and breather.
j. Excessive speed, due to light load operation and idling.
k. Lubricating oil carbon formation, usually due to improper oil, underheating and low-load operation.

REFERENCES

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Texas Company, *Diesel Engine Operation, Fuel and Lubricating Oils*, booklet published for operators, 1945.
———, *Lubrication*, Published for operators, Monthly Magazine.
Young, L. E. and C. W. Porter, *General Chemistry*, Prentice-Hall Inc., New York, 1942.

QUESTIONS

1. What technical information does an operator need on the subject of oils and lubrication, and where may he obtain additional information?

2. Should the operator depend on his own judgment when selecting and using a lubricating oil, or should he take advantage of the advice of experts?

3. What is oxidation of the lubricating oils and what causes it?

4. What factors determine the extent of oxidation?

5. What general conditions contribute to oxidation?

6. Why is it desirable to prevent emulsions of oil and water?

7. Is it possible that emulsions alone would clog the oil ducts?

8. What are the means employed to remove sludge and contaminants from the oil in the crankcase?

9. How is sludge formed as a result of oxidation?

10. Should colloidal sludges impair the lubricating properties, and foul the lubricating system when they can be clarified in the natural course of engine operation?

11. Why is it advisable to remove colloidal sludges as formed?

12. What is the usual precaution when sludges form in the crankcase?

13. What are the purposes of a filtering and reclaiming system?

14. What is an effective method of oil purification?

15. Name the kinds of sludge found in the crankcase oil, and tell how each is formed.

16. What is the composition of oil sludge?

17. What is soot, and how does it promote sludge?

18. What promotes the formation of oil-tar sludge?

19. What are "detergent" oils, "additives," and compounds?

20. What advantages are claimed for detergent oils?

21. What is the real meaning of "detergency"?

22. How does the additive reduce the sludge formation and prevent carbon deposits?

23. What determines the efficiency of detergent oils?

24. What chemical tests are made to determine the limits for changing the oil, and what factors are involved?

25. Name some of the precautions when using additive oils.

26. Why cannot chemically active filters be used with additive oils?

27. What is foaming, and what causes it?

28. Is foaming more serious with additive oils or with straight mineral oils?

29. Is it important to prevent aeration of oils? Why?

30. What is done to reduce or prevent aeration of oils?

31. What kind of filters are recommended for additive oils?

32. What are some of the filter problems?

33. What are the requirements for a satisfactory filtering system?

34. What is meant by "organic acidity," and how is it determined?

35. What is the "copper-strip test," and when is it used?

36. What may cause excessive lubricating oil consumption?

37. What checks are made for low lubricating oil pressures before adjusting the relief valve?

38. How may dilution be prevented, and what steps are taken when oil is found to be diluted?

39. Why are pressure adjustments important, and what checks are made at the time adjustment of pressure of the oil is considered?

40. What importance is attached to valve stem lubrication?

CHAPTER 12

FUEL OIL AND COMBUSTION PROBLEMS

Introduction. The chemical nature of a Diesel fuel, particularly its ignition quality, determines its performance in an engine. In the gasoline engine, it is desirable that the self-ignition point be as high as possible in order to avoid the detonation of the last portion of the charge to burn. The desirable quality of ignition in the Diesel engine is quite the opposite of that for the gasoline engine fuels. In the Diesel engine, it is highly desirable that the self-ignition point be as low as possible in order that the fuel may ignite in the shortest possible time after its injection into the combustion chamber. The designer of the Diesel engine undertakes to develop a combustion system that completely burns all the fuel in the time available without too great a rise in the combustion pressures.

Ignition quality and the delay period. It has already been shown that, for any given injection timing, the rate of pressure rise produced during the second stage of combustion depends upon the duration of the delay period in the first stage of combustion, or the time required during the first stage of combustion for the fuel to start burning. The longer this delay period, the more rapid is the rate of pressure rise, and hence the peak attained, since a greater amount of fuel will be injected before the rate of injection and burning comes under the control of the injection system. A reduction of this delay period concerns the fuel oil supplier in the same manner as it does the designer. The shorter this delay period, the longer is the second period during which the mechanical control of the injection is exercised. While the duration of the injection and the delay period are determined to a great extent by the designer, as outlined in the discussion of the fuel-mixing problems, it is also one of the principal fuel oil problems. It must be kept in mind that the ignition quality of a Diesel fuel is determined

by its chemical structure, and for this reason, many problems directly concerned are chemical problems.

Properties of Diesel fuels. Various properties of Diesel fuels must be considered from chemical and physical standpoints. The principal ones are ignition quality, viscosity, boiling range, cleanliness, and carbon test. Specifications usually list the Conradson carbon test and the sulfur, flash, and fire points. These requirements of a Diesel fuel are well established, and may be briefly defined as follows:

1. Viscosity. There are three important considerations and problems related to viscosity that have some relation to other engine operation and maintenance problems.

a. The viscosity of the fuel must be sufficiently low for the fuel to flow freely at the lowest temperature at which it will be stored and handled. Of course, in cold climates, ample provisions are made for heating the tanks and fuel lines during extremely cold periods.
b. The viscosity must be high enough, and therefore have adequate lubricating qualities, to lubricate the injection pump parts and prevent leakage at the nozzles and pumps.
c. The viscosity should be suitable for the fuel injection system itself so that it will readily atomize, as well as give correct penetration into the combustion chamber.

2. Cetane number. The cetane number must be sufficiently high to provide satisfactory starting as well as to meet the combustion requirements of the particular combustion system.

3. Gravity. The gravity of a good Diesel fuel is usually between 22 and 28 when maximum economy is concerned. This property is not as important as proper viscosity and ignition quality, since the difference in heat content has only slight variation with gravity.

4. Carbon residue. It is desirable that the carbon residue be low in order to obtain clean combustion.

5. Corrosion test. It is highly desirable that the fuel be free of corrosive properties.

6. Flash point. The legal requirements for safety in handling fuels are specified by fire insurance and underwriters codes, and all fuels meet these requirements.

7. Sediment. It is desirable that there be as little bottom sediment and water as possible, but it is not always possible

to secure absolutely clean fuel, or keep it clean in transit and storage. Cleanliness is one of the most desirable and essential properties of a Diesel fuel. More than a trace of foreign matter in the fuel will cause fuel injection pump and nozzle troubles.

Classification of Diesel fuels. Diesel engines are usually designed to operate on a given grade of fuel, depending on the design of the engine. The various grades of Diesel fuels have been classified by the American Society for Testing Materials (ASTM). Tables 12-1 and 12-2 list these classifications and the type of engine for which each grade would be most suitable. This particular classification was approved in 1941, but there have been other recommendations since that time. Considerable experimental work has been done recently, and during

TABLE 12-1
ASTM--Diesel Fuel Oil Classification

	Grade of Fuel			
	1-D	2-D	3-D	4-D
Viscosity, Saybolt Universal, @ 100° F, min	—	32.6	—	—
@ 100° F, max	—	45.5	65	140
Poor point, F°, max	0	20	35	35
Cetane number, min	50	45	35	30
Flash point, F°, min	100	140	140	140
Carbon residue, % by weight, max	—	0.20	1.0	3.5
Ash, % by weight, max	0.01	0.01	0.02	0.05
Sulfur, % by weight, max	0.05	1.0	0.5	2.0
Water and sediment, % by volume	0.05	0.05	0.1	0.5
Distillation: 90%, F° F, max	—	650	—	—
End point, F°, max	590	700	—	—

TABLE 12-2
Fuel Recommendation—Type of Engine
ASTM Data

Grade of Fuel	Type of Engine
1-D	Solid injection engines operating at over 100 rpm.
3-D	Solid injections engines operating from 350 to 1000 rpm.
4-D	Air injection engines operating from 200 to 400 rpm, and solid injection engines operating with cylinder diameters of 16 in. when operating under 240 rpm.

the war by the Army and Navy in experimental activities. It is expected that much information will soon be available on new fuel oil specifications. Report of some recent research will be quoted in this chapter.

Fuel oil specifications. Fuels are sold on specifications. The grade of fuel, and many other items in addition to those listed are included in commercial specifications. The importance attached to fuel oil specifications is evidenced by the attention given to this matter by nearly all engine builders. The manufacturers usually specify exactly what kind of fuel should be used with their engines. There is a definite relation between fuel specifications and Diesel engine operating problems that should be well understood by the operator. Fundamentally, these basic relations are as follows:

1. Distillation. Distillation is determined by boiling. The lower the boiling range, the less the smoke that is emitted by the high-speed Diesel engine. High boiling point fuels give trouble in several ways.

2. Viscosity. In addition to the lubricating value necessary for fuel pump and injector parts lubrication, viscosity determines the size of the fuel spray droplets formed by the injection nozzle. This droplet size determines the degree of atomization and the penetrating capacity of the droplet. The atomization and penetration qualities of the fuel oil spray must be determined for any fuel before it is used in the Diesel engine.

3. Ignition quality. The ability of the fuel to ignite spontaneously in a particular combustion chamber determines the ease of starting the engine; smoking and knocking also determine engine success and these are related to ignition quality. When self-ignition temperatures are too low, the engine will smoke at light loads under low temperature operation.

4. Gravity. The gravity of the fuel is related to the Btu content, this relationship being approximately:

$$\text{Btu per pound} = 17{,}680 + 60 \times \text{API gravity}.$$

Since heavier fuels have greater Btu content value per gallon, the heavier fuels are more economical. However, the lighter fuels are more desirable. Gravity, however, should not be used to govern the selection of the fuel or gauge its suitability for any particular operation, since fuels of the identical gravity may differ widely as to viscosity and ignition quality; and it is

these two characteristics that are the most important in selecting a fuel for Diesel engines.

5. *Carbon residue.* The amount of free carbon that may be expected to form in the combustion chamber and on the piston may be determined by the carbon residue tests, a standard procedure for all fuels.

6. *Ash content.* The ash remaining after a sample is burned for testing usually consists of such impurities as sand, rust, and other extremely abrasive matter. The ash content should be low to avoid wear of the engine cylinders, valves, and rings. The impurities form a large part of the ash in carbon deposits.

7. *Water and sediment.* Fuel cleanliness is necessary to prevent damage and wear of fuel pump parts. Water and sediment cause corrosion of these parts, and this is an item that should be checked for all fuel oils.

8. *Sulfur content.* Any excessive amount of corrosive sulfur will damage the engine, since the sulfurous acid will react with the water and form sulfuric acid which causes wear of the liner and other engine parts. Sulfur may also reach the crankcase through condensation of the water vapor during cold weather, and interfere with the lubrication of the engine parts and corrode the bearings and shafts.

9. *Cetane number.* This is a method of measuring the ignition quality of the fuel, accomplished by determining the ignition delay in an engine cylinder by a standard test procedure. The complete method is described by the Coordinated Fuel Research Committee in the report.

Fuel requirements of automotive Diesel engines. A progress report of the Automotive Diesel Fuels Division of the Coordinating Fuel Research Committee, Society of Automotive Engineers, was published in the *SAE Journal*, March, 1945. This paper, presented at the SAE National Fuels and Lubricants Meeting, Tulsa, Oklahoma, November 10, 1944, reported results to date of a study undertaken to determine the effects of ignition quality, viscosity, and volatility of fuels on engine performance, with particular reference to engine deposits, odor and lachrymation, low-temperature starting, power output, fuel consumption, exhaust cleanliness, and engine smoothness. This report indicated the importance of cetane number in engine starting, combustion roughness, misfiring, and varnish formation. It also showed that less viscous and correspondingly high volatility fuels gave better engine combustion, as evidenced by cleaner

exhaust and less deposition in the engines tested. For all practical purposes, this report noted, it seems that the API gravity of fuel can be taken as a measure of its heating value, which, in the opinion of these investigators, appeared to be the most important single property affecting economy and power.

Seven special test fuels were obtained by the committee for a series of tests on a group of engines, and the experiments were conducted in such a way as to permit the study of the effect of variation in a single fuel property while holding the other properties substantially constant. Three groups of fuels were used for studying the separate effects of cetane number, volatility, and viscosity. The specifications and combinations of these fuels are shown in Tables 12-3 and 12-4. Group I was selected to show the effect of cetane number; Group II, the effect of viscosity; and Group III, the effect of boiling range. An abstract of the report follows, and the accompanying graphs indicate the results obtained.

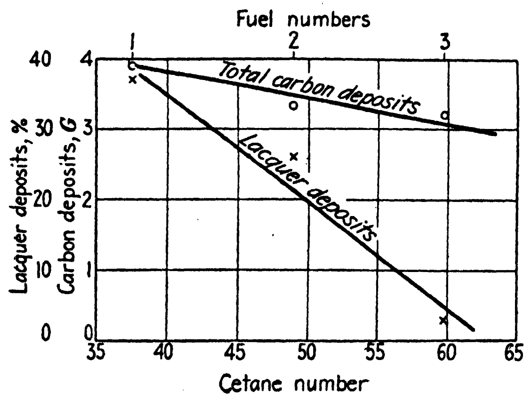

FIG. 12-1. Variation of engine deposits with cetane number (engine No. 7).

Engine deposits. The engine deposits included combustion chamber deposits of carbon and lacquer formed in the ring belt area and on the skirt of the piston. Figs. 12-1 and 12-2 show the deposits plotted against fuel properties. It is noted that a change in carbon deposits resulted in a corresponding change in lacquer formation. Both carbon and lacquer formation and deposits decreased with increasing cetane numbers, but increased with increasing viscosity.

TABLE 12-3*

PHYSICAL PROPERTIES OF CFR FULL-SCALE DIESEL TEST FUELS

Property	Fuel Number 1	2	3	4	5	6	7
Ignition quality:							
Diesel index number	40.5	56.6	70.7	64.2	43.9	53.8	48.7
Cetane number	37.5	48.9	59.7	46.6	44.3	50.5	45.2
Viscosity at 100° F:							
Kinomatic centistokes	3.09	2.98	2.865	1.378	7.387	3.324	2.705
Saybolt Universal, sec	36.1	35.9	35.4	30.4	49.8	36.98	35.2
Distillation, F:							
IBP	382	371	361	311	452	484	333
10% point	444	436	429	347	528	504	399
50% point	508	515	519	394	608	516	514
90% point	604	609	612	491	691	536	639
EP	669	654	658	543	730	558	700
Gravity, API	30.7	35.9	41.1	44.7	28.3	34.8	34.5
Flash, PM, F	169	167	158	119	226	230	147
Pour point, F	−30	−5	+15	−45	−30	−15	+5
Water and sediment	Trace	Trace	Trace	Trace	Trace	Trace	Trace
Corrosion at 212° F	Pass	Pass	Pass	Pass	Pass	Pass	Pass
Total sulfur, %	0.409	0.215	0.079	0.1003	0.103	0.105	0.26
Carbon residue (10% Btms)	0.102	0.025	0.014	0.012	0.917	0.026	0.035
Ash	0.005	None	None	None	0.035	None	Trace
Color	2½	< 2	1	¼	......	1+	2½

Fuel Combinations—With One Property Varying

Group No.	Fuel No.	Cetane No.	Gravity, API	Viscosity, SSU	Distillation F IBP	10%	50%	90%	EP
I	1	37.5 } Main variation	30.7	36.1	382	444	508	604	669
	2	48.9	35.9	35.9	371	436	515	609	654
	3	59.7	41.1	35.4	361	429	519	612	658
II	4	46.6	44.7	30.4 } Main variation	311	347	394	491	543
	2	48.9	35.9	35.9	371	436	515	609	654
	5	44.3	28.3	49.8	452	528	608	691	730
III	6	50.5	34.8	37.0	484	504	516	536	558 } Main variation
	2	48.9	35.9	35.9	371	436	515	609	654
	7	45.2	34.5	35.2	333	399	514	639	700

* *SAE Journal*, March, 1945.

TABLE 12-4*
Test Engine Specifications

Make	Company Reporting	Engine No.	Model	Bore, in.	Stroke, in.	Displacement per Cylinder cu in.	No. of Cylinders	Compression Ratio	Cycle	Rated Bhp	Compression Pressure, psi	Range of Speed, rpm	Combustion Type	Injection Pump	Nozzle	Nozzle Opening Pressure, psi
Caterpillar	Caterpillar	1	D-3400	3¾	5	55.2	4	18.5	4	32	745 at 1525	600/1650	Precombustion	Own	Own	1750
	Sinclair	2	D-4400	4¼	5½	78.0	4	17.1	4	44	695 at 1400	600/1600	Precombustion	Own	Own	1750
Fairbanks Morse & Co.	Fairbanks, Morse	3	36A4¼	4¼	6	85.1	4	15.2	4	40	540 at 1200	500/1200	Precombustion	Bosch	Bosch	1750
	Pure Oil	4	36A4¼	4¼	6	85.1	1	15.2	4	10	540 at 1200	500/1200	Precombustion	Bosch	Bosch	1750
	Tidewater	5	36A4¼	4¼	6	85.1	1	15.2	4	10	540 at 1200	500/1200	Precombustion	Bosch	Bosch	1750
General Motors	Atlantic	6	371	4¼	5	71.0	3	19.0	2	83	500	500/2000	Open	Own	Own	1500
	Shell	7	171	4¼	5	71.0	1	19.0	2	15	500	450/1200	Open	Own	Own	1500
	Tidewater	8	371	4¼	5	71.0	3	19.0	2	83	500	500/2000	Open	Own	Own	1500
Hercules	Socony-Vacuum	9	DRXB	4⅜	5¼	79.0	6	14.5	4	104	....	450/2000	Precombustion	Bosch	Bosch	1600
Mack	Mack	10	END-457	4¼	5⅜	76.1	6	14.6	4	122	375 at 165	500/2200	Lanova	Bosch	Bosch	1700
	Standard Oil, N.J.	11	ED	4⅜	5¾	86.3	6	14.57	4	131	480 at 1000	600/2000	Lanova	Bosch	Bosch	1700

* *SAE Journal*, March, 1945.

Low-temperature starting. The investigation established the relative importance of those properties of the fuels that are thought to have the greatest effect on low-temperature starting. Starting temperatures were plotted against various fuel properties, and the only definite trend indicated was the relation between the ease of starting and cetane number. This is shown in Fig. 12-3. As temperature decreased, the minimum temperature for satisfactory starting increased with an increase in cetane number.

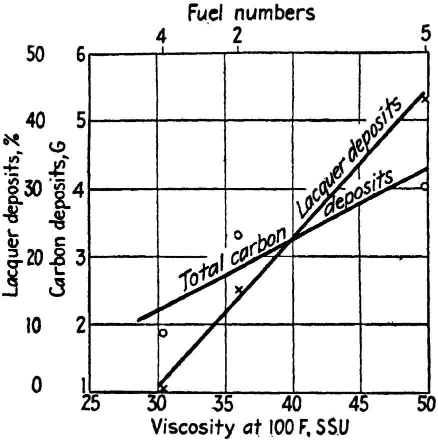

Fig. 12-2. Variation of engine deposits with viscosity (engine No. 7).

Fuel consumption. Tests at various loads and speeds to determine those properties of fuels of the seven test fuels that have the greatest influence on fuel consumption were run on various engines. The comparisons, based on heat content and/or API gravity, were reported. The difference between

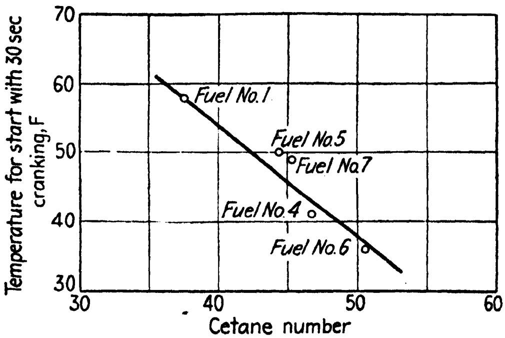

Fig. 12-3. Low-temperature starting tests (engine No. 11)—relation of ease of starting to cetane number.

fuel consumption on a weight and a volume basis is important, and is shown in Figs. 12-4, 12-5, 12-6 and 12-7. Specific fuel consumption decreased directly with an increase in API gravity for all loads and speed when expressed in pounds per brake

horsepower-hour for full rated load, as is shown in Fig. 12-4. On the volumetric basis the specific fuel consumption (pt per bhp-hr) was found to increase directly with an increase in API gravity for all loads and speeds, as is illustrated in Fig. 12-6.

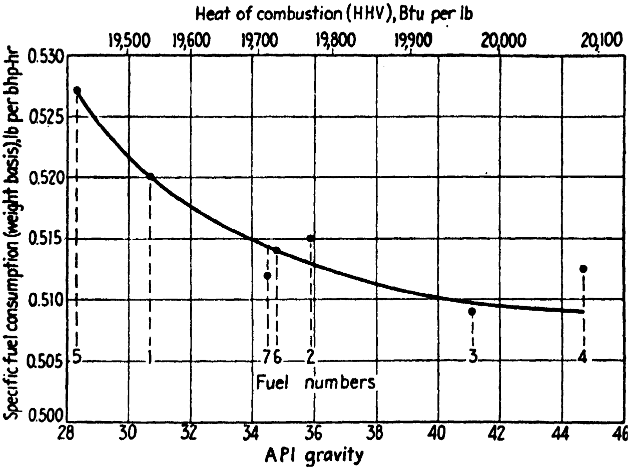

FIG. 12-4. Variation of average specific fuel consumption (weight basis) with API gravity; results are average of data obtained on engines Nos. 3 and 8 for all loads and speeds.

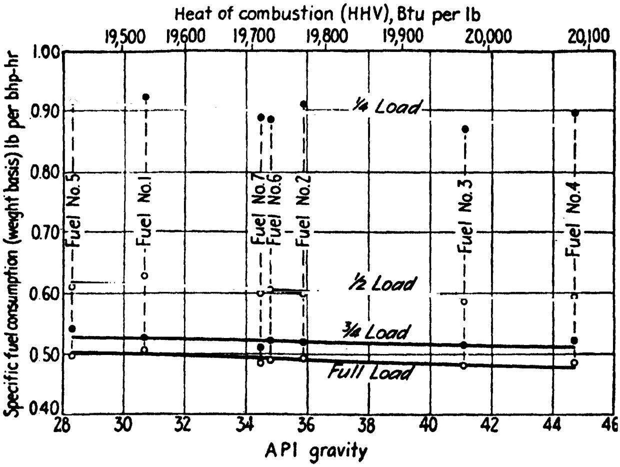

FIG. 12-5. Variation of average specific fuel consumption (weight basis) with API gravity; results are average data for engines Nos. 3 and 8 at full rated speed.

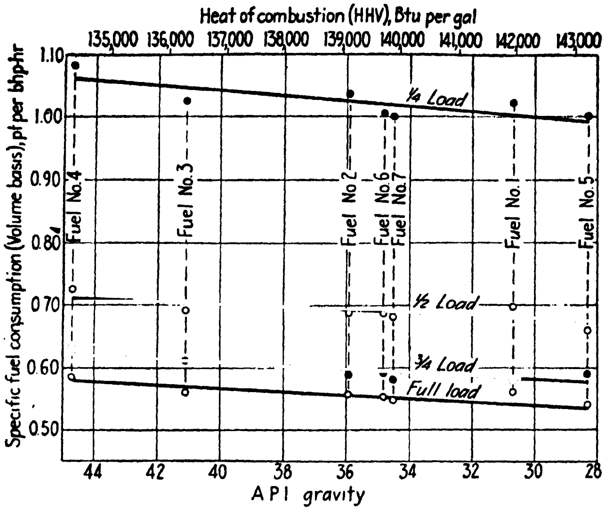

FIG. 12-6. Variation of average specific fuel consumption (volume basis) with API gravity; results are average data on engines Nos. 3 and 8 at full rated speed.

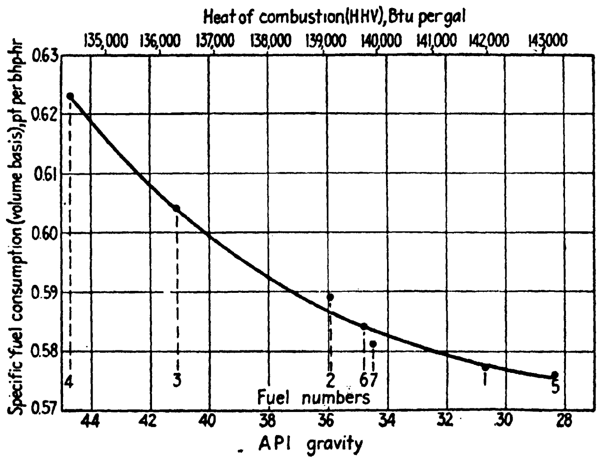

FIG. 12-7. Variation of average specific fuel consumption (volume basis) with API gravity; results are average data obtained on engines Nos. 3 and 8 for all loads and speeds.

Exhaust smoke. The average smoke value for the various fuels as it is related to fuel viscosity is shown in Fig. 12-8. There is an increase in smoking with an increase in viscosity at all engine loads. Cetane number had but small effect on

smoke under all conditions of the tests, however, the relation between smoking and mid-boiling point, Fig. 12-9, is similar to that obtained for viscosity, Fig. 12-8, which is not unexpected since viscosity and mid-boiling point vary directly with fuels of constant cetane number. Cetane number, however, is a factor in smoking at accelerated speeds after idling period, according to a previous report referred to but not quoted.

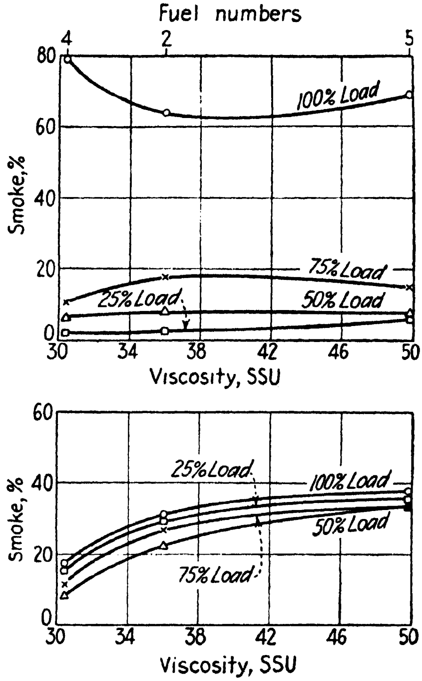

FIG. 12-8. Variation of smoke with viscosity for engine No. 8 below and engine No. 9 above. Smoke increases with increase in viscosity at all loads and speeds.

Effect of cetane number. The report pointed out that from a standpoint of relative roughness, as well as the ratio of the rates of pressure rise, engine roughness decreases with increase in cetane number. This is in line with previous experience.

Effect of viscosity and volatility. Fig. 12-10 indicates an increase in roughness as the viscosity is increased, which is said to be due to the slight increase in British thermal units per pump stroke obtained with the higher viscosity fuels. The ASTM 90% and end points showed no perceptible effect on roughness over the range studied.

As a result of this investigation and previous studies, a new set of fuel requirements or specifications was proposed. At the suggestion of several Diesel engine manufacturers and operators, the chairman of the Automotive Diesel Fuels Division of the CFR appointed a committee to review the available data and make recommendations as to the fuel requirements of the various classes of Diesel engines. The committee met and on the basis of the results of the work done by the Full-Scale

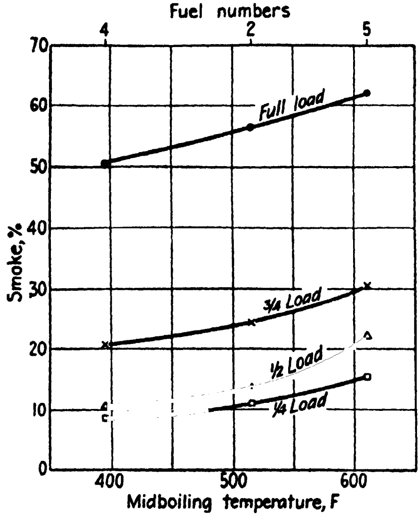

FIG. 12-9. Variation in smoke of four engines (Nos. 3, 8, 9, and 10) with mid-boiling temperatures. This graph is similar to that obtained for viscosity, since viscosity and mid-boiling point vary with fuels of constant cetane number.

Group, supplemented by the general knowledge and experience of the individual members, drew up the Diesel Fuel Classification, Table 12-4. This classification was discussed at the meetings of the Automotive Diesel Fuels Division of the SAE and the CFR, and was accepted by both groups without change. This may be compared with the ASTM classification previously given in this chapter (Table 12-1).

Viscosity and operation. It is now evident that the control of the injection timing and the amount of fuel injected depends upon keeping the viscosity of the fuel within prescribed limits. Some of the common distillate fuels used for high-speed Diesel engines have a low viscosity that may be the source of certain operating difficulties. When the viscosity is too low, a considerable leakage past the fuel pump plungers takes place as

well as inevitable wear of the injection pump and valve parts. This leaking of the pump will vary the injection timing to some extent as well as reduce the amount of fuel injected. As the wear on the injection pump elements continues, the leakage of thin fuel increases to a point where continuous small maintenance adjustments become troublesome. Such adjustments, usually made to improve the smoothness of operation and to eliminate smoky exhaust, involve the possibilities of getting

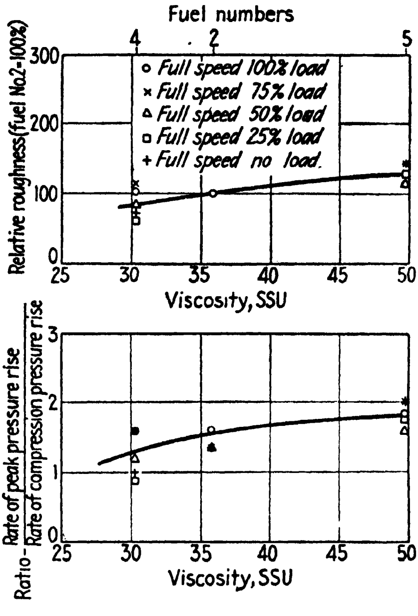

FIG. 12-10. Engine roughness at rated speed (Engine No. 11). Engine roughness at rated speed increases regardless of engine load as fuel viscosity is increased, possibly because of the slight increase in Btu's as fuel viscosity is increased.

the fuel injection system entirely out of proper adjustment. When thin fuel leaks into the cylinder, there is a tendency for the rings to gum up. The fuel may also mix with the lubricating oil, and again, it is this thin fuel oil of low viscosity that does not provide adequate lubrication of the fuel injection pump and nozzle parts; this permits friction, which increases the rate of wear, all of which, in turn, promotes a faster rate of wear.

On the other hand, when fuels are too thick—of high viscosity—there will be considerable difficulties in starting the engine in cold months. While it is usually possible to provide simple

TABLE 12-5*

FUEL REQUIREMENTS FOR DIESEL ENGINES[a]

Grade of Diesel Fuel Oil[b]	Flash Point, F[c] Min.	Pour Point, F[d]	Water and Sediment, % by Volume Max.	Carbon Residue, % by Weight Max.	Ash, % by Weight Max.	Distillation Temperatures, F: 90% Point Max.	Distillation Temperatures, F: End Point Max.	Viscosity at 100° F: Kinematic, centistokes Min.	Viscosity at 100° F: Kinematic, centistokes Max.	Viscosity at 100° F: Saybolt Universal, sec Max.	Sulfur, % by Weight[i] Max.	Corrosion	Alkali and Mineral Acid	Cetane Number Min.
No. 1-D	100 or Legal		0.05	0.05[e]	0.01	550[f]	...	....	....	....	0.5	Pass	None	40
No. 2-D	140	.	0.05	0.25[e]	0.01	650	700	2.0(32.6)[g]	6.0(45.5)[g]	...	1.0	Pass	None	45
No. 3-D	140		0.10	0.25	0.02	....	. .	. .	12(65)[g]	....	1.5	..	None	35
No. 4-D	140	.	0.50	2.00	0.10	..		. .	....	140[h]	2.0	..	None	30

[a] With the exception of the changes in the estimation of Conradson carbon, pour point, ash, 90% point, and cetane number, these requirements are the same as those listed in the Diesel Fuel Oil Classification published in the *Proceedings of the American Society for Testing Materials*, Vol. 41, 1941.

[b] No. 1-D Diesel fuel is a distillate fuel of high volatility for use in high-speed engines (speed range above 1200 rpm); No. 2-D is a distillate fuel of medium volatility for high-speed engines (speed range above 1200 rpm); No. 3-D is a distillate fuel of low volatility for medium-speed engines (speed range between 500 and 1200 rpm); and No. 4-D is a viscous fuel for low-speed engines (speed range below 500 rpm).

[c] The flash point has no bearing on performance of a fuel in an engine. However, it is required for shortage, or for legal limits.

[d] To avoid flow restrictions in cold weather, the pour point of a Diesel fuel must be specified at least 10° F below the lowest fuel temperature reached in service.

[e] On 10% residuum.

[f] Set to assure a light fuel for critical service, where smoking on heavier stock might be objectionable.

[g] Equivalent values, Saybolt Universal, seconds.

[h] Oil more viscous than No. 4-D may be specified by referring to No. 5 fuel oil (see ASTM Designation: D 396) with an agreement between the purchaser and seller regarding limits for carbon residue and sulfur.

[i] Although the information regarding the absolute maximum limits for sulfur is still inconclusive, recent evidence indicates that the sulfur permissible in these recommendations may be excessive.

* *SAE Journal*, March, 1945.

means of heating the fuels in tanks at stationary installations, such provisions may not be feasible for automotive applications. However, means of heating and warming the fuels for automotive and tractor engines are being developed for use in extremely cold climates.

Late burning. It will be remembered that the last stage of combustion is that which takes place instantly when the remainder of the charge is introduced into the engine. This is substantially what can be expected when the correct kind of fuel oil, having the proper viscosity, is used. When the fuel has an appreciable amount of heavy residue, however, there will be some late burning during the expansion stroke. The after-burning is not completed when the exhaust valve opens, and there will be a dirty exhaust. A substantial amount of half-burned, or partially burned fuel will be deposited in the combustion chamber, on the piston and cylinder walls, and eventually some of it reaches the rings, works into back of the grooves, thereby promoting ring sticking. This partly burnt fuel, which lodges on the cylinder liner, tends to form corrosive acid, which attacks the metal of the liner; and if sulfur is present, wear is rapid. A large amount of cylinder wear in the high-speed Diesel engine cylinder can be traced to such poor or incomplete combustion and accelerated corrosion. As corrosion products are scraped away at each stroke of the piston, fresh deposits are made, which result in accelerated corrosion and wear.

Heavy fuels. The fuel itself is not directly related to the cylinder wear, since the fuel can be considered to cause the wear only so far as it is related to poor or incomplete combustion. When fuels have high boiling points, the wear is greater because the heavy fractions are not completely burned, and therefore they strike the cylinder walls. When the fuel is cooled by contact with the walls of the liner, the burning is brought to an end altogether. When fuels are not completely burned by the time the droplets penetrate the combustion space, and strike the comparatively cold cylinder walls, it is because the fuel's viscosity is too high and causes a penetration too great for the size of the engine combustion chamber. The medium- and slow-speed engines can burn heavier fuels containing a larger quantity of heavy residual products without the same ill effects, because there is more time in which to burn the fuel. However, residual fuels should not be used in the medium- and

high-speed engines at the present time, because there is an ample supply of desirable fuels meeting the specifications now on the market. It sometimes happens that high viscosities result from blending heavy residual fuels with gas oils. The more the residual fuels mixed with the blend, the further is the boiling range extended. Large stationary plants may be designed and engineered to burn heavy crudes and residual fuels, but special equipment is always provided for this purpose.

Methods of burning heavy fuels. When high-sulfur fuels must be used, certain precautions are considered essential. When sulfur-bearing oils are used in the Diesel engine, the sulfur is converted to sulfuric acid or sulfur dioxide, and when oxidized, becomes sulfur trioxide. When the engine is operating at the average high temperature, the average sulfur content does not damage the engine. Such fuels having an average of 5 per cent sulfur have been burned in the 2-cycle engine. The sulfur dioxide remaining from each combustion is immediately replaced with a fresh air scavenging charge. However, when using this kind of fuel oil, the exhaust pipes were usually insulated and were made of cast iron instead of steel. The exhaust gases, upon cooling, condensed the water vapor present, formed sulfuric acid, and set up corrosive action in the pipes. Insulating the exhaust pipes is a recognized practice when the use of a fuel of more than 2 per cent of sulfur is necessary. The purpose of the insulation is to hold the heat in the exhaust pipes in order to prevent condensation in the exhaust lines at temperatures that may occur during operation. Insulated exhaust pipes are also used when burning the heavier fuels. The engines are started and stopped on lighter fuels to burn out the sulfur before shutting down the engine.

Centrifuging fuel oils. Centrifuging removes the water, mud, sand, tank scale, metallic particles, and other foreign matter usually found in the fuels. The centrifuge in larger plants has been the practice for some years. Centrifuging the fuel oil helps to reduce or eliminate various causes for sticking exhaust valves, deposits on the cylinder walls and combustion chambers, also prevents this carbon from coming in contact with the lubricating oil. Centrifuging the lubricating oil removes practically all accumulations in the crankcase oil, reduces the chances for piston seizure, leaky valves, excessive wear on the liner as well as the pump and other parts. In addition, therefore, there will be less blow-by of the combustion

gases, fewer fuel nozzle troubles, and less valve sticking. The carbonization of the nozzles ends in carbon and deposits on the piston and the rings.

In addition to the centrifuge and filtering methods, adequate screens and filter elements connected to the fuel supply system and settling tanks should be provided. The heating of the fuel before centrifuging is also an advantage, usually accomplished by the installation of hot water coils in the settling tanks, to which the cooling water from the engine is regularly circulated. When regularly heated and centrifuged before use in the engine, the fuels will be cleaned of nearly all the foreign matter. The water, sludge, and emulsions formed with water and the carbonaceous residues of half-burned oil that clog the passages and cause shut down of the engine, will be satisfactorily eliminated by careful centrifuging.

Whenever half-burned fuel oils and lubricating oil carbon, sludge, and asphaltic and carbon content from the sludge form and act as binders for the abrasive foreign matter, that is, hold it against the cylinder between the piston and the rings, there is serious grinding and lapping action that will produce a rapid wearing away of the liner and the rings. The only satisfactory solution to these problems is to keep this foreign matter out of the fuel by filtering and selection, as far as possible, of clean fuels, plus the use of the centrifuge whenever needed.

When low-grade fuel oil must be used in the Diesel engine, it is desirable from an economical standpoint that all the precautions just enumerated be observed. While better fuels cost more money, it is now the settled opinion that such fuels are more economical in the long run than the cheaper fuels with their attendant problems as outlined. It is true that restricted fuel specifications tend to make the price of Diesel fuels higher. Engines continue to be built to burn a wide variety of fuels when adjusted for them and when provisions are made for the use of heavier fuels. The problem of fuels is still a lively one for study and discussion. However, the best fuel is the cheapest fuel.

Combustion problems in the engine cylinder. The appreciable friction between the piston and the cylinder, with the lubricating surfaces influenced adversely by the high temperatures and by-products of combustion, is responsible for the wear and for the deposits on the combustion chamber. The study of the combustion problems therefore relates to what goes

on inside the engine cylinder during the combustion period. These combustion problems have been simplified by study of the combustion process and the development of the combustion chamber as well as the fuel injection system. This was previously discussed. The study of the combustion problem from the fuel standpoint has lead to certain important observations.

1. *Exhaust smoke.* The usual evidence of poor combustion of the fuel is smoke at the exhaust. When this smoke is due to the excess of lubricating oil getting into the combustion chamber and burning as fuel, the smoke is easily eliminated by eliminating the lubricating trouble. When the fuel oil is being only partly burned, the smoke will usually be black and produce a tar in the cylinder. Failure of fuel to ignite is shown by a blue smoke. Smoke at the exhaust also makes its appearance at light loads, or during temporary periods of overload. As a result of temperature drop at low loads, the fuel ignites so late that the burning is not completed by the time exhaust valves open and release the fuel only partly burned; this is what occurs upon starting the engine.

2. *Smoke limit.* The maximum power rating of the Diesel engine is usually determined by the power developed at the so-called "smoke limit." This is reached when the amount of fuel injected exceeds the limit of the air, or the required excess air. Whenever there is more fuel than can be properly burned by the air present, the "smoke limit" has been reached. This smoke is the index of unburned fuel.

Sources of smoke at the exhaust. It is essential that the operator learn to determine the source of smoke at the exhaust. The first step is to determine if the smoke comes from one or more cylinders, or if all of them are making smoke. Cylinders that are smoking will be indicated by a higher temperature on the exhaust pyrometer, as late burning is caused by conditions that accompany high temperature. Opening the ports at the exhaust valves will also show whether the smoke is coming from that particular cylinder. Operators may test for smoke by shutting off one cylinder at a time, and noting whether the smoke disappears. This should never be done when the engine is running at full load. If one cylinder is cut out when the engine is running at full power, the governor immediately opens up to permit injection of additional fuel into the other cylinders, resulting in overloading them, which in turn, causes smoking

of the cylinders as a result of the overloading. When proper fuel is being used, the chances that only one or two cylinders are smoking, and not all of them, can be taken for granted. When one or more cylinders are found to be smoking, certain adjustments can be made as listed in the instruction book.

Atomization and viscosity. When the viscosity is not correct for the engine in question, the smoke is due to the failure of the fuel to atomize properly, or it may be due to its penetration too far into the combustion chamber and striking the cylinder walls. On the other hand, when the fuel is too light, it does not penetrate the air in the combustion chamber sufficiently to become properly mixed with it, which results in incomplete combustion. When the fuel oil is too heavy, the viscous particles penetrate too far, strike the cylinder walls, and do not break up into fine enough droplets, thus fail to mix properly with the air. In any case where the fuel fails to mix properly with the air within the time available, and hence does not completely burn before the exhaust valve opens, there will be smoke at the exhaust.

It has been previously pointed out that the high temperature and turbulence of the precombustion chamber engine makes fine atomization of the fuel less necessary than in the open combustion chamber. The multiorifice injection nozzles used with open combustion chambers better atomize the fuels than do the single-hole nozzles; however, due to the smaller holes, higher injection velocities and higher injection pressures, the multiorifice injection nozzles tend to wear faster, and frequently stop up with the heavier fuels. When fuels are too light, an attempt is frequently made to employ higher injection pressures in order to increase the penetrating power of the droplets. This can be done with the single-hole orifice nozzles; but for the smaller holes, the droplets actually decrease in size and hence penetrating capacity diminishes as the injection pressure is increased.

Another difficulty encountered when using fuels that are too thin is that dribbling at the nozzle occurs in certain types of injection systems. With the common-rail system, the constant high pressure maintained on the injection valve makes it necessary to use a more viscous fuel to avoid leakage and dribbling. Higher viscosities are usually specified for these engines.

Injection difficulties. Whenever injection nozzles or valves stick, smoking is certain to occur. Sticking of the injection valve may be due to the failure of the fuel to lubricate the valve.

The clearance between the valve and the guide is extremely close, and when these parts are not properly lubricated, they first overheat, expand, and then commence to stick and gall. The first sign of this sticking is the erratic injection that causes misfiring and smoke at the exhaust. Another difficulty may be due to dirt in the fuel getting to the injection valves and causing sticking. Also, any corrosive substances in the fuel may corrode the fuel pump plunger or the injectors; this usually occurs when the combustion is generally poor in the first place. The smoke may be due to carbon deposits on the injector tips that affect the shape of the spray pattern and distort the spray formation. Sometimes, this is mistaken for incorrect injection timing. Injection timing may be changed as a result of worn cams and cam toes, and cause incomplete combustion, resulting in partly burned fuel and smoke.

Diesel knock and fuel problems. The operator's only method of reducing the combustion noise in the Diesel engine is by shortening the delay period by the means at his disposal. He can shorten the time elapsing between injection of the fuel and ignition by one or two methods:

1. A higher cetane than that being used can be secured, possibly at a higher cost. Before deciding to change the fuel grade, he should make sure that the change is needed, using a sample for a test run and observing the improvements if any.
2. Another means of shortening the delay period is to increase the compression pressure, which gives a higher temperature for ignition of the fuel. Compression pressures should be checked by means of the pressure indicator. Low compression pressure may be causing the trouble. Low compression may be due to such factors as restrictions in the air intake system, or fouled-up air filters, worn rings or cylinders, defective valves, or insufficient valve clearances. The operator should not be too ready to blame the fuel oil for all smoking he experiences with his engine.

Summary of fuel and combustion problems. The experienced operator and maintenance man can easily recognize fuel combustion problems by the evidence apparent when these problems are developing. He takes immediate steps to remedy the trouble. However, it is not always an easy matter for even an experienced man to determine what the problem is, what the cause may be, or what factors contribute most to the conditions

observed. It is frequently essential that a number of factors be considered and checked before the proper remedy can be applied. There are five groups or problems, or classifications of causes and effects, to be studied and observed in diagnosing fuel oil problems and difficulties. The fuel-mixing problems were studied in Chapter 5. With this background for consideration of the various factors involved in the design and function of the injection and combustion system, other evidence of the failure of the fuel oil or the injection system may be considered. Fuel and combustion troubles may be summed up as follows:

1. Poor combustion. As has already been shown, there are numerous reasons for poor combustion. The chief reasons, important in the order named are:

a. The load on the engine may not be balanced over all cylinders, with some of the cylinders pulling more load than others. The exhaust temperature pyrometer and the indicator are used to determine which cylinders are overloaded and which are not carrying their proper share of the load.
b. Improper injection, due to the injection valve sticking, which in turn, may be caused by the fuel being too low in viscosity. Sediment and water in the fuel also contribute to valve sticking.
c. Injection valve leaking—as a result of too low viscosity, or of pressure waves in the fuel lines.
d. Carbon in the injector, caused by the use of improper fuel, or overloaded injectors. Regrinding the fuel valve may be necessary.
e. Air in the fuel lines. This air must be bled out carefully as detailed in the instruction book.
f. Improper size nozzles, orifices too small, fuel spray striking the cylinder walls, and cold surfaces of the combustion chamber.
g. Low injection pressure, on account of weak injector valve spring. Replace the spring.
h. Incorrect injection timing—or injection not adjusted for the particular load condition, or not adjusted for the particular fuel used.
i. Insufficient air—due to restricted air intake, or incorrect valve timing, incorrect valve lift, or leaking piston rings.

j. Improper fuel—cetane number too low, improper viscosity, or high nonvolatile residue mixed with the fuel blend.
k. Low compression pressure and temperature—caused by excessive blow-by, due to worn rings, worn pistons and cylinders, or stuck rings, too light lubricating oil, and the like.
l. Compression ratio too low, valves leak, light load conditions, jacket water temperature too low—permitting engine to operate too cold, speed too low, or cold intake air.

2. Smoke at the exhaust. Smoke at the exhaust will result whenever there are the following:

a. Poor combustion, as outlined above.
b. Excessive lubricating oil consumption.
c. Misfiring, or intermittent ignition.
d. Overloading of the engine, unbalanced cylinders.
e. Uneven distribution of the load on the cylinders; regardless of the cause, this results in smoke at the exhaust.

3. Noisy operation. Diesel engines are inherently noisy, but unusual noise and knocking are usually due to:

a. Low cetane fuel. Use higher cetane fuels.
b. Low compression temperature, due to low pressure.
c. Injection timing too early.

4. Lack of power. Whenever the capacity of the engine to pull the load falls off, and the speed drops, there are certain to be some evidences of various combustion troubles, and usually fuel oil problems must be considered. Among these are:

a. Poor combustion as previously outlined.
b. Lack of sufficient air, restriction in the air intake.
c. Low heat content of the fuel used.
d. Restriction in the exhaust, such as carbon formation in the exhaust ports, or incorrect exhaust valve timing, or stuck or broken valves.
e. Exhaust valve leaking, resulting in low compression pressure.
f. Improper lubrication, resulting in tight bearings; tight pistons and piston rings may also result in reducing the capacity and power output of the engine.

5. *Combustion chamber and cylinder deposits.* One of the surest results of improper combustion and fuel problems are deposits in the combustion chambers. Such deposits are always a sign of fuel trouble, especially when the carbon is the hard, flinty type, found on the valves, piston head, and in the combustion chamber. These are due to the following as previously discussed:

a. Dirt from the intake air—dust that should be removed by the air filter. The air filter should be kept clean and should be serviced regularly.
b. Sediment from the fuel oil, and rust, which should have been kept out of the fuel, either by careful handling and straining, or for large plants, by continuous filtering or centrifuging.
c. Carbon formation—fuel carbon is different from carbon resulting from burning and baking of lubricating oil. Fuel carbon is formed by (1) use of fuel of high carbon residue, (2) poor combustion for any reason, (3) continuous overloading of the engine, and (4) misfiring, or uneven load and overload on some cylinders.
d. Dribbling of fuel nozzles—this is a prolific source of carbon on top of the piston, on the valve stems and other parts of the combustion spaces.
e. Failure of the fuel to atomize—or its penetration too far into the combustion space and striking the comparatively cool cylinder and combustion chamber walls, where it is only partly burned, after which the hot incandescent engine parts burn it into hard, flinty carbon found on the head and on the combustion chamber spaces.

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Texas Company, *Diesel Engine Operation and Maintenance—Fuel and Lubricating Oils.*

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QUESTIONS

1. How fine must the fuel be divided to obtain good combustion?
2. How is poor combustion indicated?
3. What is the effect of injection pressure on droplet size?

4. What is the effect of the size of the nozzle orifice on droplet size?

5. What other factors affect the droplet size?

6. What may cause leakage between the pump and the plunger?

7. What causes leaky nozzles, or dribbling of injection valves?

8. Is the rate of leakage increased with increase of injection pressure?

9. How can the rate of leakage be decreased?

10. How is the ignition quality, or the self-ignition temperature, of the fuel determined or measured?

11. What important factors are related to the viscosity?

12. What is cetane number, and what does it indicate about the fuel?

13. What is the importance of gravity?

14. What does foreign matter do to the fuel injection system?

15. What attention is given to and what importance is placed on fuel oil specifications by nearly all Diesel engine builders?

16. What is the effect of the boiling range of fuel oil on smoking?

17. What is carbon residue, and how is this determined by test?

18. What does thin, leaking fuel do to the piston rings?

19. What causes after-burning, and what is its effect on valves?

20. How are the fuel oil problems related to cylinder wear?

21. Why are residual fuels undesirable, and what effect does the use of these fuels have on burning when they are mixed with the fuel blend?

22. What provisions should be made for burning heavy fuels?

23. What precautions are necessary when using high-sulfur fuels?

24. Under what conditions of operation is the use of sulfur fuels safe?

25. Why are exhaust pipes made of iron instead of steel, and what is the reason for insulating the pipe when using sulfur on heavy fuels?

26. What are the advantages of centrifuging fuel oils?

27. What foreign matter is removed by the centrifuge?

28. What other facilities are employed to keep fuels clean?

29. When carbon, containing grit and dirt, forms between the piston and cylinder, what happens to the cylinder?

30. Is it desirable to use low-grade fuels in medium-speed engines?

31. What is known as the "smoke limit"?

32. What is the relation between smoke "limit" and "excess air"?

33. What is the composition of the smoke?

34. How can the course of smoke at the exhaust be detected?

35. How is smoke related to viscosity?

36. What smoking condition is observed when the oil is too thick, and why and how does a thick fuel cause smoke?

37. Name some of the fuel injection difficulties.

38. What is corrosion and what part does it play in bringing about fuel troubles?

39. What are the causes of Diesel knock, and what remedies are available to the operator to reduce it?

40. What evidence of fuel troubles does the operator look for?

41. Name the five general classes of fuel oil troubles or problems.

42. Name the usual methods available to the operator for shortening the fuel-delay period.

43. What are the usual causes of delayed ignition in the engine after it has previously been operating satisfactorily?

44. What do combustion chamber deposits indicate?

45. Name the chief causes for the loss of power, or falling off in the ability of the engine to pull its rated load.

DIESEL FUEL PROBLEMS

1. In studying the following table of fuel characteristics, ignore the slight and doubtful effects and mark pronounced effects only. Check ($\surd$) those "right" and encircle (O) those "wrong." (Approximate value for high-speed engines.)

Gravity, deg. API	16	32	48
Viscosity, Saybolt sec	20	40	80
Flash (° F)	70	140	180
Pour point (° F)	0	70	100
End point (° F)	300	575	700
Conradson carbon % of 10% bottom	0.01	0.1	1.0
Ash	0.001	0.01	0.1
Water and sediment	0.05	0.5	1.5
Sulphur	0.01	0.1	1.0
Cetane number	45	60	75
Heat value, Btu	17,000	19,000	31,000

2. Underscore the correct items in the following: Injection pumps are sensitive to: Gravity, Viscosity, Flash, Pour point, Volatility, Conradson carbon, Ash, Water and Sediment, Sulfur, Cetane number, Heat Value.

3. Check the following statement with "yes" or "no" check marks.

High cetane fuel is desirable, because the engine runs
(a) Smoother.
(b) Starts easier.
(c) Smokes less.
(d) Uses less fuel per bhp-hr.

4. Check the following statement for true or false:

A heavy engine knock, due to low cetane fuel, might be alleviated by (1) retarding the injection; (2) advancing the injection.

(5) High Conradson carbon causes carbon deposits in the engine. Right or wrong?

(6) Low flash point helps in starting. Right or wrong?

(7) Engines may use fuels containing 1, 2 or 5 per cent sulfur without damage. True or false?

APPENDIX I

METALLURGICAL GLOSSARY

Age-hardening (precipitation hardening). A process for the heat-treatment of certain nonferrous alloys to increase strength and hardness.

Annealing. Heating and cooling primarily (a) to induce softness, (b) to relieve internal stresses, (c) to obtain the optimum combination of strength and ductility, or (d) to reduce oxide. See "Stress-equalizing Annealing" and "Stress-relief Annealing."

Brinell hardness number (Bhn). A number, expressed in kilograms per square millimeter, calculated from the area of the impression surface in a specimen caused by the penetration of a 10-millimeter diameter standard hard steel ball under a load of 3000 kg acting for 5 to 30 sec, or of 500 kg acting for 30 to 60 sec. The heavier load is used for harder materials. For very hard materials a carbide ball is substituted for the steel ball.

Btu. British thermal unit. The quantity of heat necessary to raise the temperature of one pound of water by 1° F.

Charpy impact (see impact strength). The energy, in foot-pounds, absorbed in fracturing a notched specimen that has been prepared according to definite standard dimensions and is supported in a standard manner at both ends. Several forms of notch and sizes of specimen are used, and both notch and size must be specified in a given case, since variations in either will result in different unit values for the same material.

Coefficient. A number expressing the ratio of change under certain specified conditions such as temperature, length, volume, and so on. (See "Coefficients of Thermal Expansion.")

Coefficient of electrical resistivity. The fractional change in electrical resistivity per degree of temperature change, expressed in ohms per circular mil foot or michrom centimeters. It is frequently the average change in resistivity over the range 20° to 100° C (68° to 212° F) divided by the resistivity at 20° C and by 80 (the temperature differential).

Coefficient of thermal expansion. The fractional change in length of a material per degree of temperature change as compared with the length at the reference temperature, usually 0° C (32° F). It is expressed as inch per inch per deg C or F.

Compressive yield strength. The stress in compression (pushed together) at which a material exhibits a specified limiting set, commonly taken by the offset method as 0.20 per cent of the specimen's original length. Expressed as psi.

Corrosion fatigue. The endurance limit of a material when in contact with a specified corrosive medium. See "Endurance Limit."

Creep strength. The rate of continuous deformation under stress at a specified temperature. Generally expressed as psi to produce 0.1 per cent elongation in 10,000 hr at temperature indicated.

Density. The weight of a metal, usually expressed in pounds per cubic inch or grams per cubic centimeter. Do not confuse with "Specific Gravity."

Ductility. The property that permits deformation under tension without rupture. Values for "Elongation" and "Reduction of Area" generally are taken as the measure of ductility.

Elastic limit. The maximum stress, in pounds per square inch, at which a material exhibits a slight deviation from the straight line (proportional limit) but will return to the original length upon release of load.

Electrical resistivity. The resistance of a material to passage through it of an electric current. Expressed as ohms (units of resistance) per mil ft or as microhms (millionth of an ohm) per centimeter cube at a specified temperature.

Elongation. The amount of permanent stretch, after fracture in tension, expressed as percentage of the specimen's original length.

Endurance limit. A measure of the limit of safe loading for materials to be used under repeated, cyclic changes of stress. Expressed as psi. Properly, it is the maximum stress to which a metal can be subjected for indefinitely long periods without damage. In practice values are taken at a specified number of cyclic changes of stress (see "Fatigue Strength").

Fatigue strength. Usually synonymous with "Endurance Limit" but properly the stress to which a metal can be subjected for a specified number of cyclic changes of stress. Expressed as psi.

Hardness. Resistance to indentation, penetration, scratching or bending. Expressed by means of "Brinell," "Rockwell," "Scleroscope," or "Vickers" hardness numbers, depending upon the testing machine used.

Heat transfer. The passage of heat from a hot to a cold body, by conduction through intervening layers of solid, liquid, or gas. Over-all rate of heat transfer through a given system of obstructions is expressed in units of heat, per unit of area of obstructions exposed, per unit of time, per unit of difference in temperature between the hot and cold bodies (Btu per sq ft per hr per ° F). The amount of heat transferred is measured in units of heat per unit of time (Btu per hr). See "Thermal Conductivity."

Heat-treating. An operation or combination of operations involving the heating and cooling of a metal to obtain certain desirable conditions or properties, and not for the sole purpose of mechanical working.

Impact strength. A measure of toughness. The stress to fracture a notched specimen with a single blow. Expressed in foot-pounds of energy absorbed. Designated as "Charpy" or "Izod" impact strength depending on the testing machine used.

Ipy. Inches penetration per year. The average depth to which uniform corrosion would penetrate if a specimen were exposed to corrosion, on one side only, 24 hours per day for 365 days. Calculated from weight loss. See "Mdd."

Izod impact (see impact strength). The energy in foot-pounds, absorbed in fracturing a notched specimen that has been prepared to definite standard dimensions and supported in a standard manner as a cantilever. Several forms of notch and sizes of specimen are used and both must be specified in a given case since they yield different values for the same material.

Johnson's limit. The stress, expressed in pounds per square inch, at which the rate of deformation is 50 per cent greater than it is over the linear portion of the stress-strain curve.

Magnetic transformation point. The temperature at which a normally magnetic material becomes substantially nonmagnetic. Also called the Curie point.

Mdd. Milligrams per square decimeter per 24-hr day. The term for expressing average loss in weight from corrosion.

Modulus of elasticity. The ratio, within the elastic limit, of stress to the corresponding strain. Expressed in psi for four types of stress: tension, torsion, compression, shear.

Poisson's ratio. The ratio of the transverse strain to the longitudinal strain within the elastic limit determined by direct measurement. An approximate relationship is:

$$\text{Poisson's Ratio} = \frac{\text{Tensile Modulus}}{2 \times \text{Torsional Modulus} - 1}$$

Proof stress. The stress that may be applied without leaving permanent elongation of more than 0.001 inch per inch of the original length of the specimen after removal of that stress. Expressed in psi.

Proportional limit. The maximum, in psi, at which strain or deformation is directly proportional to stress.

Psi. Pounds per square inch.

Reduction of area. The difference between the original cross-sectional area of a specimen and the least cross-sectional area after rupture in tensile tests. Expressed in percentage of the original cross-sectional area.

Scleroscope hardness number. A number determined by the height of rebound of a diamond-tipped hammer dropped from a fixed height.

Shear strength. The stress required to produce fracture when impressed vertically upon the cross section of a material. Expressed in psi.

Specific gravity. The ratio of the weight of a solid or liquid to the weight of an equal volume of water.

Specific heat. The amount of heat necessary to raise the temperature of a substance by 1° F. Expressed as Btu per pound per ° F.

Stress-equalizing annealing. Heating and cooling to homogenize stresses so as to afford the best possible combination of ductility and strength.

Stress-relief annealing. Heating and cooling to effect partial softening. Also called "temper annealing."

Tensile strength. The stress required to rupture in tension (pull). Expressed in psi. Also called "breaking strength," "ultimate strength," and "ultimate tensile strength."

Tension impact. The energy, in foot-pounds, absorbed in rupturing a specially prepared specimen by allowing a moving weight to strike the specimen in such a manner that the specimen is parted suddenly in tension.

Thermal conductivity. The measure of the heat a substance will conduct through itself. Expressed in Btu, per hour, per sq ft of exposed surface, per ° F difference between the adjacent hot and cold bodies, per inch thickness (or the metric equivalents). Do not confuse with "heat transfer."

Thermal expansion. The increase in length caused by heating. Expressed in inches of increase, per inch of original length, per degrees of temperature.

Torsional properties. Figures expressing values of a material when stressed by twisting.

Torsion impact. The energy, in foot-pounds, absorbed in rupturing a specially prepared specimen by allowing a moving weight to strike the specimen in such a manner that the specimen is ruptured suddenly by torsion.

Toughness. Resistance to impact. A combination of strength and ductility.

Yield point. The stress necessary to produce an elongation under load of 0.50 per cent of the specimen's original length. Expressed as psi. Do not confuse with "yield strength."

Yield strength. The stress at which a material exhibits a specified limiting set, commonly taken by the offset method as 0.20 per cent of the specimen's original length. Expressed as psi.

APPENDIX II

TABLES AND FORMULAS

FLOW OF LUBRICATING OILS THROUGH STANDARD IRON PIPES.
REQUIRED PRESSURE PER 100 FT. OF PIPE AT GIVEN DISCHARGE

20° Baumé Gravity	½ in.		¾ in.		1 in.		1¼ in.		1½ in.	
Seconds Saybolt	Cap. Gal Min	Pres. psi	Cap. Gal Min	Pres. psi	Cap. Gal Min	Pres. psi	Cap. Gal Min	Pres. psi	Cap. Gal Min	Pres. psi
200	2	12	4	8	5	4	5	1	10	1
	4	25	8	17	10	8	10	3	20	3
	6	39	12	32	15	14	20	6	30	6
	8	53	16	52	20	23	30	13	40	10
	10	84	20	76	25	34	40	21	50	15
300	2	18	4	12	5	6	5	2	10	2
	4	38	8	26	10	12	10	4	20	5
	6	59	12	40	15	19	20	9	30	7
	8	81	16	63	20	28	30	16	40	12
	10	101	20	92	25	40	40	26	50	18
400	2	24	4	17	5	8	5	3	10	3
	4	50	8	35	10	17	10	6	20	6
	6	78	12	54	15	26	20	12	30	10
	8	107	16	75	20	35	30	19	40	15
	10	135	20	102	25	46	40	29	50	21
500	2	30	4	21	5	10	5	3	10	4
	4	63	8	43	10	21	10	7	20	8
	6	98	12	67	15	32	20	15	30	12
	8	135	16	91	20	43	30	22	40	17
	10	169	20	113	25	52	40	32	50	24
600	2	36	4	25	5	12	5	4	10	5
	4	76	8	52	10	25	10	8	20	10
	6	118	12	81	15	39	20	18	30	15
	8	162	16	110	20	52	30	27	40	20
	10	204	20	140	25	67	40	35	50	25

HEAT OF COMBUSTION OF CRUDE OILS, FUEL OILS, AND KEROSENE
(*National Bureau of Standards*)*

Gravity Degrees API	Total Heat of Combustion Btu/Lb	Btu/Gal
10	18,540	154,600
11	18,590	153,900
12	18,640	153,300
13	18,690	152,600
14	18,740	152,000
15	18,790	151,300
16	18,840	150,700
17	18,890	150,000
18	18,930	149,400
19	18,980	148,800
20	19,020	148,100
21	19,060	147,500
22	19,110	146,800
23	19,150	146,200
24	19,190	145,600
25	19,230	145,000
26	19,270	144,300
27	19,310	143,700
28	19,350	143,100
29	19,380	142,500
30	19,420	141,800
31	19,450	141,200
32	19,490	140,600
33	19,520	140,000
34	19,560	139,400
35	19,590	138,800
36	19,620	138,200
37	19,650	137,600
38	19,680	137,000
39	19,720	136,400
40	19,750	135,800
41	19,780	135,200
42	19,810	134,700
43	19,830	134,100
44	19,860	133,500
45	19,890	132,900
46	19,920	132,400
47	19,940	131,900
48	19,970	131,200
49	20,000	130,700

* From Miscellaneous Publication M97 "Thermal Properties of Petroleum Products" Published by permission of the Director, National Bureau of Standards

Conversion Tables of Equivalents for Liquids at 60°F (15.56° C.)

Degrees API	Specific Gravity	Number Pounds per U. S. Gallon	U. S. Gallons per Pound
0	1.0760	8.962	.1116
1	1.0679	8.895	.1124
2	1.0599	8.828	.1133
3	1.0520	8.762	.1141
4	1.0443	8.698	.1150
5	1.0366	8.634	.1158
6	1.0291	8.571	.1167
7	1.0217	8.509	.1175
8	1.0143	8.448	.1184
9	1.0071	8.388	.1192
10	1.0000	8.328	.1201
11	.9930	8.270	.1209
12	.9861	8.212	.1218
13	.9792	8.155	.1226
14	.9725	8 099	.1235
15	.9659	8.044	.1243
16	9593	7.989	.1252
17	9529	7.935	.1260
18	9465	7.882	.1269
19	.9402	7.830	.1277
20	.9340	7.778	.1286
21	9279	7.727	.1294
22	9218	7.676	.1303
23	.9159	7.627	.1311
24	.9100	7.578	.1320
25	9042	7.529	.1328
26	.8984	7.481	.1337
27	.8927	7.434	.1345
28	.8871	7.387	.1354
29	.8816	7.341	.3162
30	.8762	7.296	1371
31	8708	7.251	1379
32	.8654	7.206	.1388
33	.8602	7.163	.1396
34	.8550	7.119	.1405
35	.8498	7.076	.1413
36	.8448	7.034	.1422
37	.8398	6.993	.1430
38	.8348	6.951	.1439
39	.8299	6.910	.1447

CONVERSION TABLE
DEGREES CENTIGRADE TO DEGREES FAHRENHEIT

C	F	C	F	C	F	C	F	C	F
−40	−40.0	+ 5	+41.0	+40	+104.0	+175	+347	+350	+662
−38	−36.4	6	42.8	41	105.8	180	356	355	671
−36	−32.8	7	44.6	42	107.6	185	365	360	680
−34	−29.2	8	46.4	43	109.4	190	374	365	689
−32	−25.6	9	48.2	44	111.2	195	383	370	698
−30	−22.0	10	50.0	45	113.0	200	392	375	707
−28	−18.4	11	51.8	46	114.8	205	401	380	716
−26	−14.8	12	53.6	47	116.6	210	410	385	725
−24	−11.2	13	55.4	48	118.4	215	419	390	734
−22	− 7.6	14	57.2	49	120.2	220	428	395	743
−20	− 4.0	15	59.0	50	122.0	225	437	400	752
−19	− 2.2	16	60.8	55	131.0	230	446	405	761
−18	− 0.4	17	62.6	60	140.0	235	455	410	770
−17	+ 1.4	18	64.4	65	149.0	240	464	415	779
−16	3.2	19	66.2	70	158.0	245	473	420	788
−15	5.0	20	68.0	75	167.0	250	482	425	797
−14	6.8	21	69.8	80	176.0	255	491	430	806
−13	8.6	22	71.6	85	185.0	260	500	435	815
−12	10.4	23	73.4	90	194.0	265	509	440	824
−11	12.2	24	75.2	95	203.0	270	518	445	833
−10	14.0	25	77.0	100	212.0	275	527	450	842
− 9	15.8	26	78.8	105	221.0	280	536	455	851
− 8	17.6	27	80.6	110	230.0	285	545	460	860
− 7	19.4	28	82.4	115	239.0	290	554	465	869
− 6	21.2	29	84.2	120	248.0	295	563	470	878
− 5	23.0	30	86.0	125	257.0	300	572	475	887
− 4	24.8	31	87.8	130	266.0	305	581	480	896
− 3	26.6	32	89.6	135	275.0	310	590	485	905
− 2	28.4	33	91.4	140	284.0	315	599	490	914
− 1	30.2	34	93.2	145	293.0	320	608	495	923
0	32.0	35	95.0	150	302.0	325	617	500	932
+ 1	33.8	36	96.8	155	311.0	330	626	550	1022
+ 2	35.6	37	98.6	160	320.0	335	635	600	1112
+ 3	37.4	38	100.4	165	329.0	340	644	650	1202
+ 4	39.2	39	102.2	170	338.0	345	653	700	1292

Weights and Measures

5280	feet	= 1 mile
16½	feet	= 1 rod
2	yards	= 1 fathom
1.152	miles	= 1 knot
1000	millimeters	= 1 meter
100	centimeters	= 1 meter
1000	meters	= 1 kilometer
0.3937	inch	= 1 centimeter
39.37	inches	= 1 meter
25.4	millimeters	= 1 inch
.6214	miles	= 1 kilometer
144	sq in.	= 1 sq ft
4840	sq yd	= 1 acre
231	cu in.	= 1 U. S. gallon
32	ounces (volume)	= 1 U. S. quart
42	U. S. gallons	= 1 bbl
1.201	U. S. gallons	= 1 Imperial gallon
1000	cubic centimeters	= 1 liter
3785	cubic centimeters	= 1 U. S. gallon
61.023	cubic inches	= 1 liter
1.0567	U. S. quarts	= 1 liter
16	ounces (weight)	= 1 pound
2000	pounds	= 1 ton—net
2240	pounds	= 1 ton—gross
2204.6	pounds	= 1 metric ton
1000	milligrams	= 1 gram
1000	grams	= 1 kilogram
453.6	grams	= 1 pound
8.328	pounds water @ 60° F	= 1 U. S. gallon
10.02	pounds water @ 60° F	= 1 Imperial gallon
62.43	pounds water @ 60° F	= 1 cu ft
.433	lbs per sq in.	= 1 foot water
.491	lbs per sq in.	= 1 inch mercury
13.61	inches water	= 1 inch mercury
14.7	lbs per sq in.	= 1 atmosphere
1.0333	kgs per sq cm	= 1 atmosphere
.0703	kgs per sq cm	= 1 lb per sq in.
.0764	lbs air @ 60° F	= 1 cu ft
550	ft lbs per sec	= 1 horsepower
745.7	watts	= 1 horsepower
1.34	horsepower	= 1 kilowatt
778	ft lbs	= 1 Btu
2546.5	Btu	= 1 hp hour
1.8	Btu per lb	= 1 calorie per kg

FORMULAS

$$\text{Diesel Index} = \frac{\text{Aniline Point }^\circ\text{ F} \times \text{Api Gravity}}{100}$$

Cetane number (calculated)

$$= \frac{\text{Aniline Pt. }^\circ\text{ F} \times \text{Api Grav.} \times 50\%\text{ Dist. Pt. }^\circ\text{ F} \times 1.4}{100{,}000} + 11$$

$$\text{Api Gravity} = \frac{141.5}{\text{Sp Gr}} - 131.5$$

$$\text{Btu per pound} = 17680 + (60 \times \text{Api Gravity})$$

$$\text{Centigrade: }^\circ\text{ C} = (^\circ\text{ F} - 32) \times 5/9$$

$$\text{Fahrenheit: }^\circ\text{ F} = {}^\circ\text{ C} \times 9/5 + 32$$

P-V-T Relationship:

$$W = 2.7\,\frac{PV}{T}$$

$$T_2 = T_1 R^{n-1}$$

$$P_2 = P_1 R^n$$

T_1 = Absolute temperature ° F at beginning of compression
T_2 = " " " " end of compression
P_1 = " pressure lb/sq in. at beginning of compression
P_2 = " " " " end of compression
R = Compression ratio
n = 1.3 for Diesel compression
W = Weight air in pounds
P = Pressure in lb/sq in.
V = Volume in cu ft
T = Abs. Temp., ° F

Brake Horsepower:

$$BHP = \frac{PLAN}{33000}$$

BHP = Brake horsepower
P = Mean effective pressure, lb/sq in.
L = Stroke feet
A = Effective area of piston—sq in.
N = Number single effective strokes per minute

CHART FOR APPROXIMATING CETANE* NUMBER FROM VISCOSITY AND GRAVITY OF FUEL
(Moore and Kaye)

Viscosity (Saybolt Universal) Seconds at 100° F	API Gravity																
	18	20	22	24	26	27	28	29	30	31	32	33	34	35	36	37	38
34	21	25	29	33	37	39	40	42	44	46	47	49	51	52	54	55	57
35	22	26	30	34	38	40	42	44	46	48	49	51	53	54	56	57	59
36	23	27	31	35	39	41	43	45	47	49	51	52	54	56	58	59	61
37	24	28	32	36	40	42	44	46	48	50	52	53	55	57	59	60	62
38	24	29	33	37	41	43	45	47	49	51	53	54	56	58	60	61	63
39	25	29	33	37	41	43	45	47	49	51	53	55	57	59	61	62	64
40	25	30	34	38	42	44	46	48	50	52	54	56	58	60	62	63	65
42	26	30	35	39	43	45	47	49	51	53	55	57	59	61	63	64	66
44	26	31	35	40	44	46	48	50	52	54	56	58	60	62	64	65	67
46	27	32	36	41	45	47	49	51	53	55	57	59	61	63	65	66	68
48	27	32	36	41	45	47	49	51	53	55	57	59	61	63	65	67	68
50	28	33	37	42	46	48	50	52	54	56	58	60	62	64	66	68	69
60	29	34	39	43	47	50	52	54	56	58	60	62	64	66	68	69	70
80	31	35	40	45	50	52	54	56	58	60	63	65	67	69	71	72	..
100	32	37	42	47	52	54	56	58	60	62	65	67	69	71	..	..	..
150	34	39	44	49	54	56	58	60	63	65	67	69	72	..	..	..	..
200	35	40	45	50	55	58	60	62	65	67	69	71	..	..	..	..	..
300	36	41	47	52	57	60	62	65	67	69	72	..	..	..	..	..	..
400	37	42	48	53	59	61	64	66	68	71	..	..	..	..	..	..	..
500	38	43	49	55	60	63	65	67	70	..	..	..	..	..	..	..	..

* Cetane numbers only slightly lower.

ASTM Detailed Requirements for Fuel Oils (1944) (Abbreviated)

Grade of Fuel Oil	Flash Point, ° F		Pour Point, ° F	Water and Sediment, per cent by volume	Carbon Residue, per cent by weight	ASH, per cent by weight	Distillation Temperatures, ° F				Viscosity, Sec.				Sulfur, %
							10 per cent Point	90 per cent Point		End Point	Saybolt Universal (at 100° F)		Saybolt Furol (at 122° F)		(Optional)
	Min.	Max.	Min.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.	
No. 1 A distillate oil for use in burners requiring a volatile fuel.	100 or legal	165	0	Trace	0.05*	....	410	...	...	560	..	..	...	..	0.5
No. 2 A distillate oil for use in burners requiring a moderately volatile fuel.	110 or legal	190	10	0.05	0.25*	....	440	600	...	...	..	..	...	..	0.5
No. 3 A distillate oil for use in burners requiring low-viscosity fuel.	110 or legal	230	20	0.10	0.15	....	...	675	600	...	45	..	...	..	0.75
No. 5 An oil for use in burners requiring a medium-viscosity fuel.	130 or legal	...	..	1.00	.....	0.10	...	...	...	...	..	50	50	..	....
No. 6 An oil for use in burners equipped with preheaters permitting a high-viscosity fuel.	150	...	..	2.00	.....	....	...	...	...	...	..	..	300	45	....

* On 10% residuum.

BEARING METALS LISTED ACCORDING TO HARDNESS

Bearing Material	Hardness Brinell		Approximate Composition, %					Corrosion* Resistance
Soft Bearings:	70°	300°	Tin	Lead	Copper	Antimony	Other	
Pure lead	4	3	..	100	....	....	....	C
Lead-indium coatings	4	3	..	96	....	....	4 In	C
Lead-tin coatings	7	4	4	96	....	....	....	B
Moraine coating	17	5	4	92	....	4	....	B
Lead-base babbitt	19	6	5	80	....	15	....	B
Lead-base babbitt	20	6	6	84	....	10	....	B
Lead-base babbitt	22	7	10	75	....	15	....	B
Tin-base babbitt	23	7	89	.....	3.5	7.5	....	A
Copper-hardened lead	22	9	5	88	1	5	1 As	B
Silver-hardened lead	23	9	2	77.8	0.2	15	5 Ag	B
Arsenic-hardened lead	21	10	1	82.5	0.5	15	1 As	B
Calcium-hardened lead	22	11	1	98	....	....	0.5 Ca 0.5 Other	C
Cadmium-nickel	33	13	..	.....	....	....	98.7 Cd, 1.3 Ni	C
Cadmium-silver	38	13	..	.....	0.5	....	97.5 Cd, 2 Ag	C
Hard Bearings:								
Copper-lead	24	20	..	30	70	....	....	C
Copper-lead	28	25	..	25	75	....	....	C
Silver	25	25	..	3	....	....	97 Ag	A
Copper-lead	42	38	3	25	72	....	....	C
Lead-bronze	62	56	10	15	75	....	....	A
Lead bronze	64	57	10	10	80	....	....	A
Aluminum	72	65	7	.....	....	....	93 Al	A
Lead-bronze	78	73	10	5	85	....	....	A

Ag—Silver, As—Arsenic, Cd—Cadmium, Cu—Copper, In—Indium, Ni—Nickel, Pb—Lead, Sb—Antimony, Sn—Tin.

NOTE: This hardness ranking should not be considered a quality rating, since for some engines soft metals are best suited, whereas for others hard bearings are superior. Also, two different bearing metals may have the same hardness, but one may be vastly better in service. Skill in manufacturing and accuracy in machining are most important in determining the quality of a bearing. Hardness values are averages of various published data.

* A—Noncorrosive.
B—Noncorrosive except under extreme conditions.
C—Requires use of noncorrosive oil.

APPENDIX III

LUBRICATING OIL SPECIFICATIONS

NAVY HEAVY-DUTY DIESEL LUBRICATING OIL SPECIFICATIONS

U. S. Navy, Bureau of Ships, Ad Interim Specification 14-0-B (INT)

Requirements of lubricating oils for Navy Diesel Engines (condensed). Issued May 1, 1941 (reprinted from *SAE Journal*, page 284, July, 1942).

DETAIL REQUIREMENTS
DIESEL ENGINE LUBRICATING OILS

Symbol..........	9170	9250	9370
Viscosity (Saybolt Universal), sec at 130° F.	140–200	220–280	320–430
Flash point (min.), ° F	350	370	400
Pour point (max.), ° F	0	10	15
Carbon residue (max.) (ash free), %	0 9	1 1	1 3
Neutralization no. (max) . .	0 5	0 5	0 5
Precipitation no. (max)	None	None	None
Corrosion	None	None	None
Ash (max.), %..	0 6	0 6	0 6

GENERAL REQUIREMENTS OF N.S. 9000 SERIES OILS AND CORRESPONDING TEST PROCEDURES:

REQUIREMENTS	TEST PROCEDURE
(1) Shall provide satisfactory lubrication for all engine parts and generator bearings of naval Diesel engines.	Tests in laboratory, submarine-type and other naval Diesel engines. Tests in Navy work factor machine.
(2) Shall be noncorrosive to bearings and engine parts.	Tests in laboratory, submarine-type and other naval Diesel engines. Underwood oxidation and corrosion test. Tests in Navy work factor machine.
(3) Shall not cause ring sticking or clogging of oil channels.	Tests in laboratory, submarine-type and other naval Diesel engines.
(4) Shall maintain minimum piston ring and cylinder liner wear.	Tests in laboratory, submarine-type and other naval Diesel engines.
(5) Quality shall not be adversely affected by mechanical or fibrous type filters or by centrifugal purification.	Tests in laboratory, submarine-type and other naval Diesel engines and special filter tests. Centrifuge tests of new oil with and without fresh and sea water.
(6) Addition of new oil to used Diesel engine oil shall not cause sludging.	Make blends of equal volume of new and used oils, shake for twenty minutes, centrifuge in tube for one hour.
(7) Shall not cause excessive carbon deposits on any part of engine.	Tests in laboratory, submarine-type and other naval Diesel engines.
(8) Each grade shall be satisfactory in all types of Diesel engines ordinarily requiring that grade oil.	Tests in laboratory, submarine-type and other naval Diesel engines.
(9) Mixtures of additive and straight mineral oils shall perform as well as the straight mineral oil alone.	Three hundred-hour laboratory Diesel engine tests on equal volumes of contract 3065 and additive oil.

(10) Additive oil shall not be affected adversely by 2% by volume of sea or fresh water.	Three hundred-hour laboratory Diesel engine test on oil plus 2% by volume of synthetic sea water (2% by volume added each 25 hours). Measure equal volumes (50 ml) of new oil and synthetic sea water, and new oil and distilled water in centrifuge tubes; shake for 20 minutes, centrifuge in tubes for 20 minutes. Determine % of stable emulsion and loss of additive.
(11) Additive agents shall remain uniformly distributed throughout the oil at temperatures from 10° F above the pour point up to 250° F.	Tentative homogeneity tests: (a) 50 ml sample dried at 212 to 220° F for three hours; cooled to 10° F below pour point; observe at 55 to 65° F. (b) Heat 150 ml sample at 250° F for three hours; observe for separation, thickening or cloud formation.
(12) Diesel engine lubricating oil shall show superiority over straight N.S. mineral oils where run in laboratory Diesel engines for at least 250 hour test periods and when used in service.	Tests in laboratory, submarine-type and other naval Diesel engines.
(13) Shall be compatible with all other Diesel engine lubricating oils previously procured.	Blend equal volumes of new oils (50 ml of each); shake for twenty minutes; centrifuge in tube for one hour.

SPECIFICATIONS FOR 11 DIESEL ENGINES USED IN NAVY TESTS OF HEAVY-DUTY LUBRICATING OIL:

Designation	Cylinders	Cycles	Rpm	Bhp
Atlas-Lanova	1	4	1800	5
General Motors, 1-71	1	2	1200	15
Caterpillar, 1 Cyl	1	4	1000	19.8
Winton X-1/20 1A	1	2	750	82.4
Caterpillar D3400	4	4	1200	22.8
Winton V-12/201A	12	2	750	900
Cummins, H	6	4	1600	95
General Motors, 6-71	6	2	1800	150
General Motors, 268A	8	2	1300	......
General Motors 16-248A	16	2	800	2000
Fairbanks-Morse 38-D8-⅛	9	2	900	2000

ARMY HEAVY-DUTY ENGINE OIL SPECIFICATIONS

Oil, Engine, for Use in Automotive Gasoline and Diesel Engines U. S. Army Specification No. 2-104B (Condensed)

(Reprinted from *National Petroleum News*, P. R300, July 7, 1943)

D. GENERAL REQUIREMENTS

D-1. Engine oil shall be noncorrosive to bearings and engine parts, shall not cause or permit piston ring sticking or clogging of oil channels and shall minimize cylinder and ring wear. This engine oil shall provide satisfactory lubrication of high-speed, automotive-type gasoline, Diesel, or spark ignition fuel engines when operated under all conditions of service. Additive agents, if used, shall not appreciably increase the tendency of the base oil to foam.

D-2. Additive agents, if used, shall remain uniformly distributed throughout the oil at all temperatures above the pour point up to 250° F. If the oil is cooled below its pour point, it shall regain its homogeneity on standing at a temperature of not more than 10° F above the pour point of the oil.

E. DETAIL REQUIREMENTS

E-2. Engine oils covered by this specification shall be in accordance with the following:

Test	Test Limits		
	Grade SAE 10	Grade SAE 30	Grade SAE 50
Viscosity, Saybolt Universal Seconds at 130° F	90 to less than 120	185 to less than 255	....
Seconds at 210°F	....	....	95 to less than 105
Viscosity Index, minimum	85	55	75
Pour Point, ° F, maximum	minus 10	0	15
Pour Point after dilution with 20 per cent Precipitation Naphtha ° F, maximum	minus 40	minus 40	....
Flash Point ° F, minimum	360	390	400

F. METHODS OF TESTS

F-3. Qualification Tests. All engine oils procured under this specification shall be tested for qualification as follows:

Required when the finished engine oil uses an additive not previously used in an engine oil qualified by the Ordnance Department.	Required when the finished engine oil uses an additive previously used in an engine oil qualified by the Ordnance Department.
SAE 10 GRADE	SAE 10 GRADE
1. Chevrolet 36-hr Oxidation Test	1. Chevrolet 36-hr Oxidation Test
2. Caterpillar Test No. 1-A	2. Caterpillar Test No. 1-A
3. Caterpillar Test No. 2-A	3. Caterpillar Test No. 2-A
4. Caterpillar Test No. 3-A	
SAE 30 GRADE	SAE 30 GRADE
1. G. M. Diesel 500-hr Test	1. Chevrolet 36-hr Oxidation Test
2. Chevrolet 36-hr Oxidation Test	2. Caterpillar Test No. 1-A
3. Caterpillar Test No. 1-A	3. Caterpillar Test No. 2-A
4. Caterpillar Test No. 2-A	
5. Caterpillar Test No. 3-A	
SAE 50 GRADE	SAE 50 GRADE
1. Chevrolet 36-hr Oxidation Test	1. Chevrolet 36-hr Oxidation Test

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